Naval Surface Warfare Center
Carderock Division
West Bethesda, MD 20817-5700

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Survivability, Structures, and Materials Directorate

Technical Report

Fatigue Design Guidance for Surface Ships

by

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1. Reference (a) requested the Naval Surface Warfare Center, Carderock Division (NSWCCD) to formulate design guidance for surface ship structures against fatigue crack initiation. Enclosure (1) presents results of an effort to calibrate a fatigue design procedure for surface ships by benchmarking different types of U.S. Navy ships that have successfully completed their service lives without experiencing fatigue crack initiation. Regardless of where or how the ships were operated, a "design" operating scenario in the North Atlantic for a limited number of days was chosen for the fatigue analyses. The design scenario, at a limited number of days, is assumed to be equivalent to the actual lifetime operating conditions. Four ship types were chosen for analysis; combatants (destroyers and cruisers), amphibious assault ships, auxiliaries, and aircraft carriers. Three ships of each type were analyzed. Results are presented in terms of permissible stress levels and stress concentration factors. Assuming future ships are of the same form and will be operated in the same way as those analyzed, this design guidance should result in a ship that will not experience crack initiation during its service life. In the event future ships are different from those analyzed, a more general procedure is outlined.

2. Comments or questions may be referred to Mr. Michael W. Sieve, NAVSEA Code 05P; telephone 703-602-5515 ext 114; email, SieveMW@navsea.navy.mil, or Dr. David P. Kihl, NSWCCD, Code 653; telephone 301-227-1956; email, KihlDP@nswccd.navy.mil.

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14. ABSTRACT
This document contains guidance to design surface ship structures against fatigue failure. In engineering terms, fatigue describes the initiation and growth of cracks and associated damage accumulation which may occur under cyclic loads and their effect on the service life of a structural member. Ships are subjected to fatigue cycles primarily by the action of seawater waves. The guidance provided in this report applies to naval monohull ships of conventional hull form, construction and material. For conventional hull forms, the procedure provides maximum permissible primary design stresses as a function of ship type and structural detail. General guidance is provided for other non-conventional ships. The permissible stresses are based on benchmark analyses of different types of U.S. Navy surface combatanats that have successfully completed their service lives without experiencing fatigue crack initiation. A "design" operating scenario in the North Atlantic was chosen for the analyses. For the fatigue damage calculations, a limited number of days is determined to be equivalent to the actual lifetime operating conditions which result in a fatigue life equal to the service life. Results are presented in terms of permissible stress levels and stress concentration factors.

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Contents

Page

Figures v

Tables v

Administrative Information vi

Introduction 1

- Fatigue Design Needs 1
- Design Philosophy 3
- Reliability-based Design 4
- Applicability and Limitations 5
- Assumptions 5

General Design Procedure 6

- Service and Extreme Loads 7
  - Spectral Analysis 8
  - Sea Spectra 9
  - Assessment of Lifetime Loads 9
- Hull Girder Stress Analysis 11
  - Stress Types and Definitions 11
  - Hull-Girder Cross Section and Strength Properties 13
  - Longitudinal Stress Distribution 13
  - Stress Concentrations 13
- Strength of Details 13
  - The Characteristic S-N Approach 14
  - The Fracture Mechanics Approach 15
- Design Method and Criteria 16
  - Design Method 16
  - Design Criteria 18

Cursory Design Procedure 20

- Lifetime Bending Moments Amidships 21
- Longitudinal Distribution of Bending Moments 22

Contents

Page

Stress Analysis 28
Hull-Girder Section and Strength Properties 29
Longitudinal Stress Distribution 29
Stress Concentrations 29
Peak Stress Range 30
Fatigue Damage Accumulation 30
S-N Curves 30
Representative Details 31
Design Criteria 33
Nominal Stresses 33
Permissible Stress Concentration Factors 34
References 35
Appendix A – Methods for Calculating Stress Concentration Factors (SCF) for Openings A-1
Appendix B – Methods for Calculating Stress Concentration Factors (SCF) Due to Misalignments B-1
Appendix C – Catalog of Structural Details C-1
Appendix D – Example Computations D-1
Acronyms, Symbols, Definitions and Conversion Factors E-1

Figures

Page

Figure 1. Block Diagram for Conversion of Data from Temporal Domain to Spectral Domain and Vice Versa 9
Figure 2. Operational Envelope for Ships 9
Figure 3. *S-N* Approaches for Fatigue Strength Assessment (Niemi 1995) 12
Figure 4. Comparison between the Characteristic *S-N* curve and Fracture Mechanics Approach 14
Figure 5. Response Histogram for Fatigue 16
Figure 6. Design for Fatigue Based on Nominal Stress 17
Figure 7. Reliability-based Design and Analysis for Fatigue 19
Figure 8. Longitudinal Distribution Factor (*DF*) 23
Figure 9. AASHTO Fatigue Design *S-N* Curves 32

Tables

Page

Table 1. Maximum Lifetime Wave-induced plus Whipping Bending Moment Coefficients 22
Table 2. Longitudinal Distribution of Maximum Bending Moment Range 23
Table 3. Speed and Heading Probabilities for Frigates, Cruisers and Destroyers 24
Table 4. Speed and Heading Probabilities for Auxiliaries, Tankers and Slow Cargo Vessels 25
Table 5. Speed and Heading Probabilities for Amphibious Assault and Fast Cargo Vessels 26
Table 6. Speed and Heading Probabilities for Aircraft Carriers 27
Table 7. Ship Characteristics and Operational Baseline Parameters 28
Table 8. Wave Height Probabilities 28
Table 9. *S-N* Curve Coefficients (Stress Ranges in ksi) 31
Table 10. Design Details and Category 32
Table 11. Permissible Stress Ranges (ksi) 33
Table 12. Maximum Permissible Stress Concentration Factor 34

Administrative Information

The work described in this report was performed by the Structures and Composites Department, Code 65, of the Survivability, Structures, and Materials Directorate at the Naval Surface Warfare Center, Carderock Division (NSWCCD) under the sponsorship of the Naval Sea Systems Command (SEA 05R). SEA 05P provided technical guidance. A portion of this work was performed at the University of Maryland (College Park, Maryland) and BMA Engineering, Inc. under contract N0016794M1227. This report is submitted in partial fulfillment of Milestone V2, Subtask IID of the Reliability Based Structural Design Program PE063564N, Project S2036-01.

Introduction

This document contains design guidance to optimize surface ship structures against fatigue failure. It applies to naval monohull ships of conventional hull form, construction and material, including frigates, destroyers, cruisers, amphibians, auxiliaries, and carriers. For conventional hull forms, the procedure provides maximum allowable primary design stresses as a function of ship type and structural detail. General guidance is provided for other non-conventional ships.

Fatigue Design Needs

In recent years, a great deal of attention has been focused on general fatigue cracking of structural details. Attention to this phenomenon is so vital that structural engineers must consider fatigue strength in their designs for those structural components that are exposed to cyclic loading. The term "fatigue" is commonly used in engineering to describe the formation and growth of cracks that may occur under repeated-loads and the effect of these cracks on the strength of a structural member. The exact mechanism of a fatigue failure is complex and is not completely understood. Failure by fatigue is evidenced by progressive cracking, which, unless detected and remedied, can lead to a catastrophic failure. When a repeated load is large enough to cause a fatigue crack, the crack will start at the point of maximum stress. This maximum stress is usually due to a geometric stress concentration (stress raiser) or a defect in the plating or weld. After a fatigue crack has initiated at some microscopic or macroscopic level of stress concentration, the crack itself can act as an additional stress raiser causing further and more rapid crack propagation. The crack grows with each repetition of the load until the effective cross section is reduced to such an extent that the remaining portion will fail with the next application of the load by yielding or compressive instability. For a fatigue crack to grow to such an extent to cause failure, thousands or even millions of stress applications may be required. The actual number of stress cycles required to cause failure depends on the magnitude(s) of the applied stress, type of the material used, and on other related factors. Rather than using an exact number, several specimens are tested and the cycles to failure are more typically described in statistical terms, such as mean and standard deviation. Cycles to failure may also be described in probabilistic terms by fitting the cycles to a probability distribution. A given number of cycles to failure may then be associated with a probability of failure.

Fatigue must be considered in the design of all structural and machine components that are subjected to repeated or fluctuating loads. During the useful life of a structural member, the number of loading cycles, which may be expected, varies tremendously. For example, a beam supporting a crane may be loaded as many as 2,000,000 times in 25 years, while an automobile crankshaft might be loaded 5,000,000,000 times in 200,000 miles.

The number of loading cycles required to cause failure of a structural component through cyclic successive loading and reverse loading may be determined experimentally for any given maximum stress level. Test results of fatigue strength of specimens are commonly reported for each test in terms of the maximum stress range, S, against the number of cycles, N. These test data are usually plotted on log-log or semi-log paper, and the resulting plot is referred to as an S-N curve. As the magnitude of the maximum stress range decreases, the number of cycles to cause failure increases. Also, under constant amplitude stresses, these curves tend to be approximately horizontal lines as a lower limit is approached. When the stress level for a specimen reaches this limit, the specimen does not fail and it is said to have reached the endurance limit (fatigue limit). The endurance limit is then defined as a theoretical stress for which failure does not take place even for a large number of loading cycles. The endurance limit for most engineering material is significantly less than the yield strength. For a low carbon structural steel, the endurance limit is about half of the ultimate strength of the steel. However, most structural applications are composed of welded connections. In welded structures, fatigue cracks generally initiate at the toe of the weld in the heat-affected zone. As with baseplate, a similar endurance limit effect occurs in weldments under constant amplitude loads. The endurance limit, however, is generally ignored in conservative designs.

Fatigue characteristics of various welded details are usually determined at room temperature, mostly in air, and sometimes in various corrosive environments. Geometry and environment can play a significant role in influencing the fatigue properties of structural details. For example, in applications in or near seawater, or in other applications where high level of corrosion is expected, reductions in fatigue life of over 50% may be anticipated. Also, since fatigue failure may be initiated at any crack or imperfection, the service condition and weld quality and profile of a specimen have a vital effect on the value of the behavior observed in the test.

The inherent nature of fatigue tests gives rise to a great deal of scatter in the data. For example, if several specimens that have been carefully fabricated, are tested at the same stress level, it is not unusual to have a variation of 10 to 20 percent in their fatigue life measured in terms of the number of loading cycles at which the specimens fail. It therefore requires several specimen tests to correctly identify a mean S-N curve for a detail, and many more to quantify the scatter about the mean S/N curve.

Fatigue cracking of structural details in ship and offshore steel structures due to cyclic loading has gained considerable attention in the past few years. Numerous research works have been conducted in this field on both the theoretical and practical aspects. Consequently, a large number of papers have been published on various topics relating to fatigue assessment and prediction. In these papers, the macroscopic behavior of materials, as well as models describing cracking, are investigated. Due to the extreme complexity in modeling the process of material cracking at the microscopic level, solutions from the microscopic aspect are rarely available or are not practically feasible. This is mainly due to the complexity of the damage process under cyclic loading and the scatter of material properties and other variables that affect fatigue behavior.

Ship and offshore structures are subjected to fatigue primarily due to the action of seawater waves and the sea environment in general. The load cycles in such an environment can be in the order of millions of cycles per year and composed of a wide range of magnitudes. Fatigue failures in ship and offshore structures can take place at sites of high stress concentration that can be classified into two major categories: (1) baseplate and (2) weldments. The former includes locations of high stress concentration such as openings, sharp re-entry corners, and plate edges.

The latter includes details associated with the miles of transverse, longitudinal and vertical welds that connect the various pieces of a ship hull. In general, the mechanisms behind these failures are described by the general approaches to fatigue life prediction as discussed in this report.

There are two major approaches for evaluating fatigue life prediction: (1) the *S-N* curve approach and (2) the fracture mechanic (FM) approach. The *S-N* approach is based on experimental measurement of fatigue life in terms of cycles to failure for different loading levels as discussed previously. On the other hand, the fracture mechanic (FM) approach is based on the existence of an initial crack in a loaded structure. This guidance is based on the *S-N* approach. The FM approach will be reserved for hull girder damage assessment in future guidance.

Traditionally, longitudinal strength has been determined by balancing the ship on a static wave. This approach has been widely accepted as an expedient means of simplifying a time dependent dynamic situation into a simple static analysis. The ability to meet operational requirements using a static balance method is implicitly based on the historical success of the method. The standard wave height used by the United States Navy (USN) in this procedure is $1.1\sqrt{LBP}$, where *LBP* is the length between perpendiculars in feet, and 1.1 is an empirical coefficient. The ship is balanced on the trough of a trochoidally shaped wave, resulting in a sagging design condition and then on the crest, resulting in the hogging design condition. The length of the wave is taken to be the same as the ship length between perpendiculars. Longitudinal bending moments and shears are then determined from the weight and buoyancy distributions, treating the ship as a free-free beam. Typically a ship is divided into 20 stations between the forward and aft perpendiculars. Cross sectional beam properties and primary stresses are determined for each station. To simplify design calculations a stress envelope is assumed taking the design primary stress limit value as constant throughout some portion of mid-body length dictated by judgment. Fore and aft, the design primary stress tapers to zero at the perpendiculars. The calculated stress must be below the design stress by a certain stress factor (margin) to account for future growth in displacement. This stress factor varies from 0.5 tsi to 1.0 tsi depending on ship type and material. The calculated primary stress cannot exceed the design stress values; otherwise additional material must be added to lower the hull girder stresses. The design primary stress limits, which vary from 8.5 tsi to 10.5 tsi, depending on material, are based on past experience and are empirical in nature. Indirectly, they could provide a check on fatigue. The need to address fatigue explicitly has become increasingly important due to desire to increase primary stress levels and to extend ship lives. Both of these trends increase fatigue damage.

Design Philosophy

The USN is currently undertaking an effort to revise the design criteria used for surface ships. The criteria (Naval Sea Systems Command 1976) have previously been presented in the form of Design Data Sheets (DDSs). DDSs have evolved over the past several decades and have been updated periodically as new technology and procedures have been developed. DDSs are based on deterministic methods and first principles, used in conjunction with empirical experience, experimental data and conservative engineering practices. Although generally successful over a wide range of applications, the DDS approach does not allow for risk and reliability assessments because it does not consider the probabilistic aspects of reliability-based ship design. In some cases, the actual factors of safety against certain failure modes are not explicitly defined and the overall reliability of the ship or structural component is unknown.

On its way to developing a fully probabilistic design approach, the USN has developed procedures that specifically address certain failure modes. These procedures, along with applicable DDSs, are currently being used to develop Load and Resistance Factor Design (LRFD) rules. The LRFD rules loosely resemble the DDS approach, but through the use of partial safety factors allow the ship to be designed to a given target reliability.

This particular guidance outlines a procedure for explicitly designing a surface ship against failure by fatigue. The maximum and cumulative lifetime bending moments replace the bending moments based on the $1.1\sqrt{LBP}$ wave. The maximum permissible stress range replaces the design primary stress envelope. Eventually the guidance will be revised to a reliability-based format, such as the LRFD format, and ultimately to a fully probabilistic approach. Until that time, it is intended to allow the ship designer to produce a ship that will successfully complete its service life before experiencing fatigue cracking.

Previously, ship design against failure by fatigue has been implicit. Using limiting primary design stress levels associated with particular types of steel, the hull would be balanced on a trochoidal wave of height equal to $1.1\sqrt{LBP}$ to determine the design hog and sag longitudinal vertical bending moments. Assuming the ship acts as a free-free beam, it would be allowed to heave and trim to achieve equilibrium between weight and buoyancy. Typically, bending moments would be calculated at twenty equally spaced stations along the length of the hull between the forward and aft perpendiculars. By proportioning the scantlings to maintain the minimum section moduli required from the design, primary stress and the design bending moments would be limited and fatigue would presumably be avoided. This method basically works well, but since the margins against failure are not explicitly quantified, the resulting ship design may be unnecessarily robust. In weight-critical applications, this approach is unacceptable. Also, since the fatigue design is not explicit, the resulting structure may be under-designed and experience fatigue cracking during its service life.

Reliability-based Design

A methodology for the development of surface ship structure reliability-based design criteria was recently constructed by the Structures and Composites Department at NSWCCD (Ayyub 1994; Ayyub and Assakkaf 1998; Ayyub and Assakkaf 1999a; Ayyub and Assakkaf 1999b; Ayyub et al. 1998; Ayyub, Beach, and Packard 1995). It was developed based on the consideration of the following three components: (1) loads, (2) structural strength, and (3) methods of reliability analysis.

The reliability-based design approach requires the definition of a set of target reliability levels. These levels can be set based on implied levels in the currently used design practice with some calibration, or based on cost-benefit analysis. The reliability-based design procedure starts with defining performance functions that correspond to limit states for modes of failure. A generalized form for the performance function for a structure is given by

$$g = R - L \tag{1}$$

where R = strength and L = loads on the structure. The failure in this case is defined in the region where:

$$g < 0.0 \text{ or } R < L \tag{2}$$

Due to the variability in both strength and loads, that is, both strength and load are defined by probability distributions, there is always a probability of failure where the distributions overlap that can be defined as

$$p_f = P(g < 0.0) = P(R < L) \tag{3}$$

Applicability and Limitations

This guidance contains two procedures that are called the general procedure and the cursory procedure. The more rigorous general procedure was used to develop the simpler and empirically based cursory procedure.

The general procedures described in this document apply to traditional ship types and configurations. It is assumed that the primary loads can be predicted with reasonable accuracy and that the fatigue strength of the fabricated material can be characterized and predicted. Basically, a lifetime histogram of applied stresses is required and Miner's cumulative fatigue damage hypothesis is used to calculate the fatigue damage. The fatigue life is thereby determined explicitly.

The cursory procedure contained within this document is based on analysis of several types of ships that have successfully completed their service life without experiencing fatigue problems. Determination of a lifetime stress histogram and explicit fatigue analysis is not required. Given the overall dimensions and type of ship, a maximum permissible design stress is provided for general categories of structural details. The maximum lifetime load magnitude and longitudinal distribution are estimated with simple algorithms and the section modulus thus defined. The ships include frigates, cruisers, destroyers, auxiliaries, amphibious assault, and aircraft carriers. They are all monohulls of conventional shape with orthogonally stiffened plate construction using ordinary and high strength steels. This procedure may not produce an acceptable design for ships that deviate from these characteristics. In such cases, the general procedure must be used and some assumptions must be made to design the ship against fatigue. If the ship configuration or construction materials are significantly different and load and fatigue strength behavior are not available, some model and fatigue tests are required. If a ship being designed is similar to any of those ships that served as the basis of this procedure, then the resulting hull is expected to have adequate strength against fatigue cracking during its service life. In general, the design allowable stresses and design conditions stated in this procedure are expected to result in a ship that has a 2.3% probability of failure during its service life under a loading condition as defined by the underlying stress-range histogram. The allowable stresses were developed according to the following requirements: (1) spectral analysis of wave loads, (2) building on conventional codes, (3) nominal strength and load values, and (4) achieving implicit reliability levels. They are applicable to frigates, destroyers, cruisers, auxiliaries, amphibious assault, and aircraft carriers for a typical 30- to 50-year service life.

Assumptions

The fatigue design procedures described in this guidance are based on the following assumptions:

1. Miner's cumulative fatigue damage rule accurately predicts fatigue damage accumulation.
2. The fatigue strength is primarily a function of applied far field stress range.
3. Mean stress effects can be ignored.
4. The fatigue behavior of a structural detail tested in a laboratory reflects the behavior of a similar detail of interest located within the ship structure.
5. The fatigue behavior can be represented by a power law function (i.e., linear in log-log space)
6. Constant amplitude endurance limit effects can be ignored when applied to variable amplitude situations.
7. Probability of failure can be represented by a normal distribution in log-log space that has constant standard deviation with stress range. The stress range versus number of load cycles to failure (*S-N*) curves associated with other than mean probability of failure plot parallel to the mean *S-N* curve.
8. The ship is a linear system, such that loads can be superposed.
9. Longitudinal vertical bending loads which act on the hull are the primary source of fatigue damage; contributions due lateral bending and to situation specific loadings such as aircraft landing or local pressure variations due to wave passage may need to be dealt with separately.
10. The allowable stress levels obtained empirically under the cursory procedure assume the ship being designed is geometrically similar and will be operated in a similar way as the ships analyzed for developing the respective allowable stress levels.

Experiments have shown the assumptions stated above to be reasonable and representative of typical ship structure. These assumptions also reflect common practice currently employed in civil engineering design codes and other industries.

General Design Procedure

Implementation of any fatigue design procedure involves:

1. the determination of the loads that are expected to act on the structure during its service life;
2. analysis to determine how the external loadings develop internal stresses at the point of interest within the structure;

3. the choice of a suitable *S-N* curve, which reflects the fatigue strength of the structural detail of interest and the desired probability of failure, are required; and
4. a fatigue damage accumulation relationship to assess the criticality of the applied stress cycles when compared to the stress cycles to cause failure; that is, the fatigue damage, and hence the fatigue life.

The general design procedure, in theory, is applicable to mononhulls (beyond and including those used to develop the cursory procedure), SWATH ships, and non-steel structure. As such, the general fatigue design procedure is outlined and described in this section.

Service and Extreme Loads

Fatigue design requires the assessment of a lifetime vertical bending moment histogram for a ship. The lifetime bending moment histogram summarizes the ranges of bending moment magnitudes (i.e., hog to sag variations) and their corresponding number of cycles expected during the ships service life. These bending moments include those due to changes in wave height and slam induced whipping. The computational process is based on spectral analysis that accounts for sea conditions, operational profile of the ship, ship characteristics, and service life. The resulting bending moments, calculated amidships, are then distributed longitudinally along the full length of the ship. Showing favorable agreement with experimental data, the longitudinal distribution is often assumed to follow that of a "one minus cosine curve". Stresses at the point of interest are then calculated from the external loads. In some cases, the effects of lateral bending and/or torsional loadings may need to be included when computing the lifetime stress histogram. Fatigue damage is primarily dependent on applied stress (and therefore bending moment) *range*. Mean stress is a secondary effect; it is therefore not necessary to include the still water contribution.

The loads of interest on a hull girder arise from the ship responding to an active seaway. Responses to these loads can be measured directly on a ship as it operates in various sea conditions. Typically however, the ship may not yet exist and estimates of load must be made. Under known conditions of heading and speed, the loads can be quantified as a function of frequency and normalized by the wave height. In this normalized form, the loads can be estimated for any known wave condition. Loads expressed in this normalized form are referred to as Response Amplitude Operators (RAOs). Algorithms have been developed from model tests and full-scale at-sea ship trials to estimate RAOs for given heading and speed combinations, provided the principle ship dimensions are known (Sikora 1998; Sikora and Beach 1986). Alternatively, one could determine RAOs experimentally from model tests and full-scale trials, or analytically from ship motion computer programs. The former method can be measured directly on the actual ship or a representative model. The latter method requires specific information about the ship, such as offsets to define the hull shape and the mass distribution throughout the hull to define inertial properties.

An operational scenario is constructed to define the anticipated conditions in which the ship is expected to operate. The operational requirements document (ORD) and ship specifications should be consulted for information on where and how the ship is expected to operate, its operability and service life before constructing the operational scenario. The operational scenario is comprised of the heading, speed and time the ship will operate in various wave conditions throughout its service life. The wave conditions are defined by existing formulations known as wave spectra. These formulations express the wave heights as a function of frequency, and are categorized by the significant wave height, or the average wave height an experienced observer would estimate visually. The RAOs and wave height spectra are used to establish the cyclic load distribution associated with the time spent operating at each speed, heading and wave height combination in the operational profile. A common assumption made during this operation is that the peak values of the cyclic load responses are represented by a Rayleigh probability distribution and the cyclic frequency of the loads occurs predominately at the average frequency of encounter between the ship and waves. From these individual cyclic load distributions, a master histogram of applied load cycles is developed which represents the entire magnitude and number of cycles the ship is expected to experience during its service life.

The lifetime fatigue loads should include wave-induced and dynamic effects, and they do not include stillwater or hydrostatic loads as they do not affect stress ranges. For some ship types, components and locations, load cycles due to lateral and torsional moments, wave slap and passage at the waterline, fluid sloshing and loads associated with internal tanks, and equipment vibration need to be considered and accounted for in fatigue design. For deck edges, longitudinal, vertical and lateral bending moments should be considered. Limited fatigue life calculations have shown the accumulated fatigue damage can range from being almost twice to being almost comparable to the damage accumulation due to vertical bending alone, depending on the phasing between the two loadings.

Spectral Analysis

A wave spectrum is a linear mathematical representation of the sea waves for a given sea state. Spectral representation of ocean waves is used in this analysis. This representation can be achieved through an auto-correlation function that fulfills all of the requirements for defining a Fourier transform. Accordingly, the auto-correlation function and its Fourier inverse for a stationary random process, where t = time, are respectively given by

$$S(\omega)=\frac{1}{2\pi}\int_{-\infty}^{\infty}R(\tau)\cos\omega\tau\, d\pi \tag{4}$$

and

$$R(\tau)=\int_{-\infty}^{\infty}S(\omega)\cos\omega\tau\, d\omega \tag{5}$$

where $S(\omega)$ = spectral density function, ω = wave frequency, $R(\tau)$ = correlation function for a random process at any instant, and τ = time interval for two times t_1 and t_2, that is, $\tau = t_2 - t_1$. The auto-correlation function can be reduced to the following expression when $\tau = 0$:

$$R(0)=E(X^2)=\int_{-\infty}^{\infty}S(\omega)d\omega \tag{6}$$

where $E(X^2)$ = mean square, and $R(0)$ in length unit squared. Figure 1 shows a block diagram for the conversion of data from temporal domain to spectral domain and vice versa.

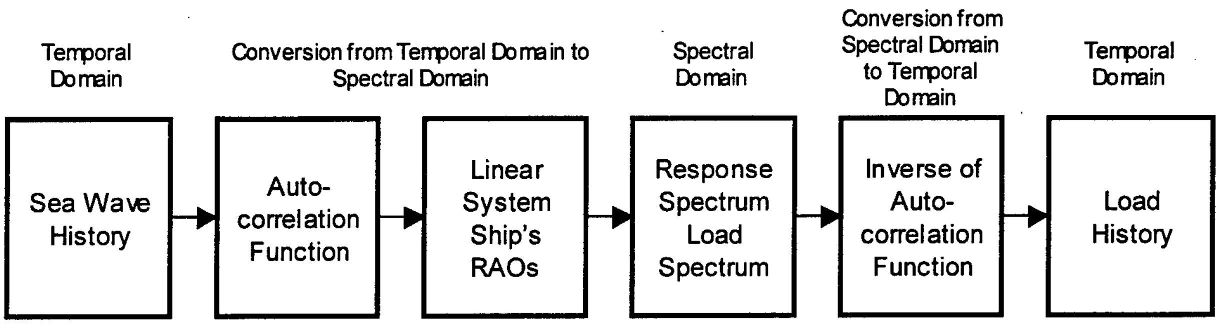

Figure 1. Block Diagram for Conversion of Data from Temporal Domain to Spectral Domain and Vice Versa

Sea Spectra

Sea spectra are used to statistically represent the surface of the sea. They indicate the amount of energy associated with a given wave frequency. The wave spectral density function $S(\omega)$ has units of wave height^2/unit of circular frequency (ft^2-sec). There are many different sea spectra in use for marine structures; each has its own merit. Probably the most common two sea spectra used for naval structures are (1) the Two-parameter Wave Spectrum and (2) the Ochi Six-parameter Family of Wave Spectra (Ochi 1978). Ayyub and Assakkaf (1999a; 1999b) describe these two kinds of sea spectra.

Assessment of Lifetime Loads

Spectral analysis is used to develop a wave-induced lifetime fatigue and extreme loads histogram by considering the operational conditions of a ship in the sea to be divided into different operation modes according to the combinations of speeds, headings, and waves heights as shown in Figure 2. Those operating conditions prone to producing slamming are further analyzed to produce slam-induced whipping loads that are then combined with the wave-induced loads.

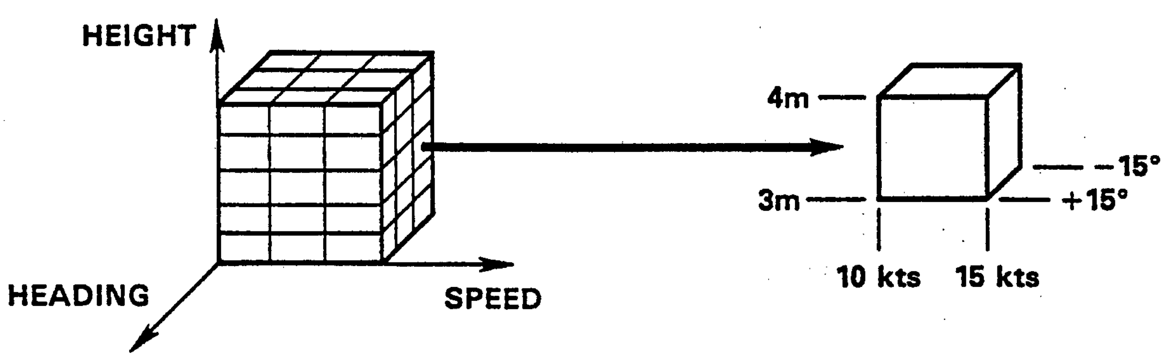

Figure 2. Operational Envelope for Ships

This procedure has been automated in a window's based computer program called SPECTRA. Sikora (1998; 1983) and Sikora and Beach (1986) documented the program; Michaelson (2000) produced a user's manual. The operation mode is defined by the ship speed, heading relative to the sea, and the sea condition. Each operation mode results in a response spectrum (in this case, the wave-induced bending moment), which is the product of wave spectrum and the response amplitude operators (RAOs), where the RAO is defined as the response per input wave height squared as a function of wave frequency. The area under each response spectrum defines a Rayleigh probability distribution assuming the response to be narrowband Gaussian. The total number of cycles included in this distribution is the product of the time of that incremental mode and the average encounter frequency.

The time spent at sea for each operational condition is given by

$$T_i = T_y P_1 P_2 P_3 P_4 \tag{7}$$

where T_y = life-time at sea; P_1 = ship heading probability; P_2 = ship speed probability; P_3 = wave height probability; and P_4 = wave spectral probability. The average encounter frequency may be determined from the second moment of the response spectrum as given by

$$\omega_{ie} = \sqrt{\frac{\sum_j A_j \omega_{iej}^2}{\sum_j A_j}} \tag{8}$$

where A_j = area under an increment of the response function, ω_{iej} = wave excited frequency of the ship at the i-th mode and the j-th response function, and N_i = the number of cycles at the i-th mode and is given by

$$N_i = \frac{T_i \omega_{ie}}{2\pi} \tag{9}$$

Examples of speed and heading probabilities, wave height probabilities and ship characteristics needed to perform the analysis can be found in the next section where the cursory design guidelines are discussed.

Slam-induced whipping responses are then determined using empirically based algorithms obtained from full-scale trials and model tests. The algorithms define the probability density function for the high-frequency whipping response associated with each operational mode. Once defined, the high-frequency whipping distribution can be combined with the low-frequency Rayleigh distribution of the wave-induced response.

The combined bending moments CM_{hog} and CM_{sag}, can be determined from the following expressions as a function of the wave induced vertical bending moments, M_{hog} and M_{sag}, the lateral bending moment, M_{lat}, and the peak-to-peak slam moment, $W_{p\text{-}p}$.

$$CM_{hog} = M_{hog} + SW + 0.5W_{p-p}\left[\exp(-\delta\ R(\Phi + 180)/360)\right] \tag{10a}$$

$$CM_{sag} = M_{sag} + SW + 0.75W_{p-p}\left[\exp\left(-\delta\ R(\Phi + 180)/360\right)\right] \quad (10b)$$

$$CM_{lat} = M_{lat} + 0.5W_{p-p} \quad (10c)$$

where SW is the still-water bending moment, R is the ratio of the natural frequency to the wave encounter frequency, δ is the log decrement of the hydrodynamic damping of the hull, and Φ is the phase angle from the initiation of the slamming event to the peak of the next wave-induced sag cycle.

The probability distribution for the wave-induced bending moments is represented by a Rayleigh distribution and the slam-induced bending moments are represented by an exponential distribution. For those cells that include slam induced whipping, slam rates and whipping magnitudes are used to estimate the number and distribution of whipping cycles that occur within each cell. The combined wave plus slam-induced bending moments are then generated on a cycle-by-cycle basis as described above. Choosing a bending moment level and calculating the number of cycles that exceed that bending moment level for each cell of the operational profile will obtain the lifetime bending moment exceedance spectrum. This step is repeated for each cell of the operational profile and the results combined to produce the lifetime exceedance curve. For fatigue analysis, a lifetime bending moment histogram can be readily constructed from the lifetime exceedance curve.

To determine the once-in-a-lifetime extreme bending moment values, an iterative procedure is employed to determine the bending moments exceeded once. Each cell is assumed to contribute fractions of a cycle toward the one complete lifetime cycle. Within each cell, the difference between the exponential and Rayleigh distributed cycles at a single cycle is assumed to be constant addition to the Rayleigh distribution below one complete cycle. Iteration produces the once-in-a-lifetime bending moment exceeded, by fractional contributions of all cells, exactly one time.

Hull Girder Stress Analysis

Stress Types and Definitions

The choice of an appropriate stress history is an important factor in the design for fatigue. The question is not really how to determine the stress history; rather, what constitutes an appropriate stress history. Using the terminology adapted by the International Institute of Welding (IIW) in 1996, the following four different approaches are classified for stress determination for fatigue design and analysis: (1) the nominal stress approach, (2) the hot spot stress approach, (3) the notch stress approach, and (4) the notch strain approach. Figure 3 shows a schematic of these approaches. Except for the nominal stress approach, the rest are commonly called local stress approaches. Probably the most common approaches for determining fatigue stresses in marine industry are the nominal stress and the hot spot approaches.

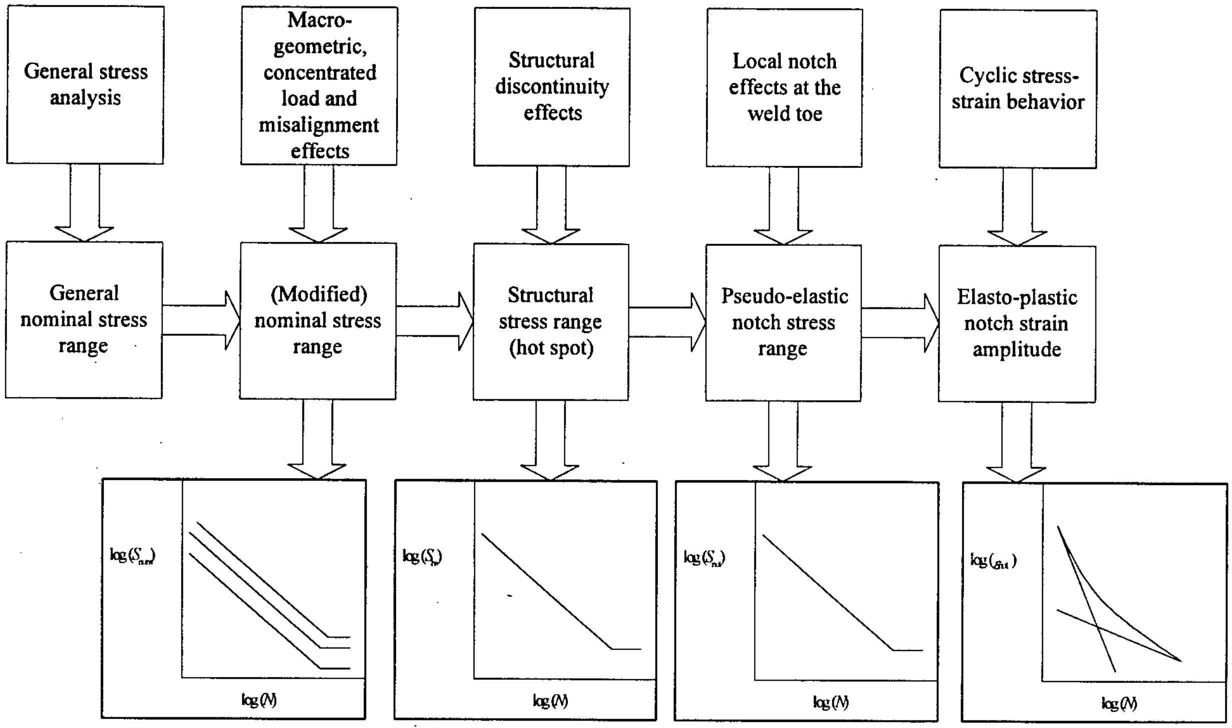

Figure 3. *S-N* Approaches for Fatigue Strength Assessment (Niemi 1995)

Nominal stress is the simplest of the four approaches. In this approach, stress is represented by an average loading of the whole structural detail under study. The nominal stress is the maximum stress due to sectional forces or moments or the combination of the two at the location of possible cracking site in the detail. In this approach, neither the weld toe nor the properties of the material constitutive relations are taken into consideration. The *S-N* curve resulting from this analysis is unique to the structural detail for which it is established. It is possible to apply one such curve for a range of similar details if there is insignificant variation in their geometry. Most current design codes divide various structural details into different classes and provide standard *S-N* curve for each class.

The hot spot stress is defined as the fatigue stress at the toe of the weld, where the stress concentration is the highest and where fatigue cracking is likely to initiate. The hot spot stress is comprised of membrane and bending shell stress parts which are linearly distributed over the plate thickness. The hot spot stress analysis takes into account two factors: (1) the local increase in membrane stress due to complex structural geometry of the welded joint and (2) the information of shell bending stress due to eccentricity. The exact weld toe geometry and nonlinear stress peak due to local notch at the weld toe are disregarded. The hot spot stress is an average nominal stress of the stresses near the weld. The advantage of the hot spot stress method is that only one universal *S-N* curve is required to define fatigue strength for all welds, if such curve exists. The disadvantage is that this approach requires a detailed finite element analysis to determine the hot spot stress.

This guidance described in this document is based on the nominal stress approach.

Hull-Girder Cross Section and Strength Properties

The cross-sectional properties of hull girders shall be defined by using the inertia sections developed in the Longitudinal Strength Drawing (Bureau of Ships 1950). The cross-sectional properties are required for stations 1 through 19.

Longitudinal Stress Distribution

The longitudinal bending moments, calculated amidships, can be distributed along the length of the hull for assessments at other stations. A "one minus cosine" function has been found to agree well with test data and is typically used to establish the longitudinal distribution. The nominal longitudinal stress-range distribution can be calculated using finite element analysis or by dividing the longitudinal bending moment range for each station by the section modulus for that station. For most longitudinal structure, a finite element model is unnecessary, and "hand" calculations are sufficient to determine the nominal stress levels.

Stress Concentrations

Global stress concentrations shall be considered as they increase stress ranges for details above the nominal stress levels. Global stress concentrations are typically associated with large changes in geometry and structural discontinuities such as deck openings, superstructure terminations, plating and member misalignment, and deck knuckles. A local stress concentration need not be considered because its effects are inherent in the *S-N* curve for the joint. Finite element analysis can be used to assess global stress concentration (Basu, Kirkhope, and Srinivasan 1996). Also, approximate methods can be used as provided in the Cursory Design Procedure, page 20.

Attaching the superstructure to the strength deck requires special consideration at the fore and aft ends. Finite element method (FEM) analysis is required to quantify biaxial stress effects and ensure smooth transition of loads into the strength deck and superstructure. The Ship Structure Committee Report 387 (Basu, Kirkhope, and Srinivasan 1996) provides a method for evaluating finite element models, results, and FEA software.

In areas of stress concentration, peak stress range is critical to fatigue life. A peak stress range is defined as the product of the nominal stress range and an appropriate stress concentration factor. Appendixes A and B provide stress concentration values for openings and misalignments, respectively.

Strength of Details

There are generally two major technical approaches for fatigue life assessment of welded joints: (1) the characteristic *S-N* approach, and (2) the fracture mechanics approach. Both of these approaches are provided herein, although only the *S-N* approach shall be used for design. The fracture mechanics approach is provided for damage assessments.

Although welded ship structure contains many different types of structural details, it is often both more efficient and prudent to design the hull to the fatigue strength of the most severe "critical detail" that is prevalent in primary structure. The critical detail will end up controlling the design since the presence of stronger details will not change the fatigue life of the ship.

Local problem areas can be addressed separately with insert plates and reinforcement to reduce the nominal stresses to acceptable levels.

The Characteristic S-N *Approach*

The characteristic *S-N* approach is based on fatigue test data (*S-N* curves) and on the assumption that fatigue damage accumulation is a linear phenomenon (Miner's rule). According to Miner's rule, the total fatigue life under a variety of stress ranges is the weighted sum of the individual lives at constant stress range *S* as given by the *S-N* curves, with each being weighted according to fractional exposure to that level of stress range. Upon crack initiation, cracks propagate based on the fracture mechanics concept as shown in Figure 4.

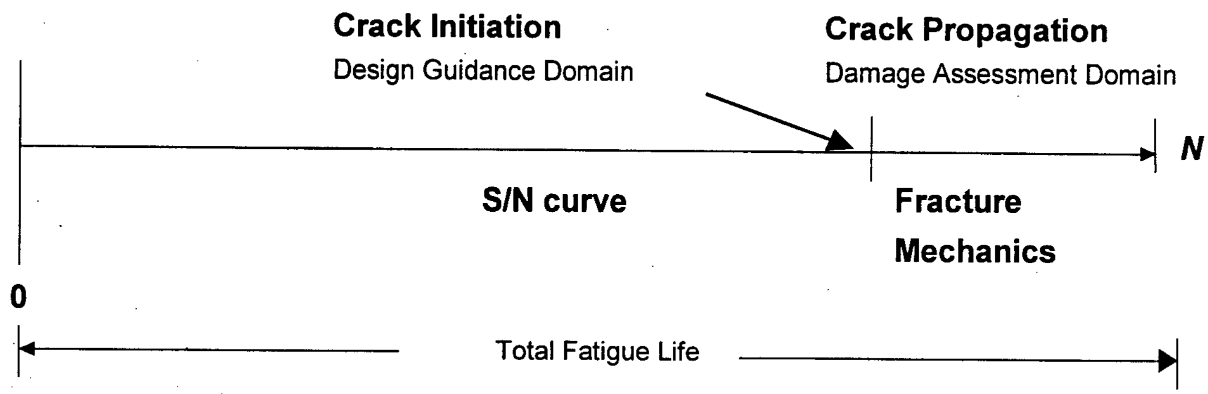

Figure 4. Comparison between the Characteristic *S-N* curve and Fracture Mechanics Approach

Fatigue life strength prediction assessment based on the *S-N* approach and stress concentration can be expressed as:

$$N_i = A k_S^b S_i^b \tag{11}$$

or

$$\log(N_i) = \log(A) + b\left[\log(k_S) + \log(S_i)\right] \tag{12}$$

where log(.) is to the base 10, and

A = intercept of the *S-N* curve

b = slope of the log-log *S-N* curve

S = stress range

S_i = stress range of the *ith* stress-range block of a stress-range histogram

k_S = fatigue stress concentration, or uncertainty, factor

N_i = fatigue life, or number of loading cycles expected during the life of a detail due to S_i

The stress-range histogram is based on a model that formulates sea state spectra, information on ship's routes and operating characteristics, and the ship behavior in an active seaway to provide a detailed history of stress ranges over the service life of the ship. The fatigue strength of various welded steel details can readily be found in many civil design codes and technical documents available in the open literature as discussed by Kihl (1999). Data for welded aluminum and glass-reinforced plastic (GRP) are also available in the open literature, but are less prevalent than that for steel. In some cases, testing may be required to develop the S/N curve specifically for a particular detail.

The Fracture Mechanics Approach

The fracture mechanics approach is based on crack growth data. For welded joints, it is assumed that the initiation phase is either negligible or used up and a flaw or crack is present just under the threshold of detection and that life can be predicted using the fracture mechanics method. The fracture mechanics approach is more detailed and it involves examining crack growth and determining the number of load cycles that are needed for small cracks and initial defects to grow into cracks large enough to cause fractures. The growth rate is proportional to the stress range. It is expressed in terms of a stress intensity factor K, which accounts for the magnitude of the stress, current crack size, and weld and joint details. The basic equation that governs crack growth is given by

$$\frac{da}{dN} = C(\Delta K)^m \tag{13}$$

where

a = crack size

N = number of fatigue cycles (fatigue life)

ΔK = $SY(a)\sqrt{\pi a}$, range of stress intensity factor

S = applied stress range

C, m = crack propagation parameters

Y(a) = function of crack geometry

Fatigue life prediction based on the fracture mechanics approach shall be computed according to the following equation:

$$N = \frac{1}{CS^m} \int_{a_0}^{a} \frac{da}{Y^m} \tag{14}$$

where a_0 = initial crack size. Equation 14 involves a variety of sources of uncertainty. The crack propagation parameter C in Equation 14 is treated as random variable; however, in more sophisticated models, Equation 14 is treated as a stochastic differential equation and C is allowed to vary during the crack growth process.

Design Method and Criteria

Design Method

The stress-range histogram at some location of interest can be developed as described in the Hull Girder Stress Analysis Section, page 11, and the Strength of Details Section, page 13, Sections. Figure 5 provides a schematic for the procedures involved for producing a moment response histogram as well as a stress response histogram. The overall design method based on meeting a design criterion is provided in Figure 6 based on limiting the fatigue damage ratio (Δ) to 1. An initial section modulus estimate must be made to start procedure.

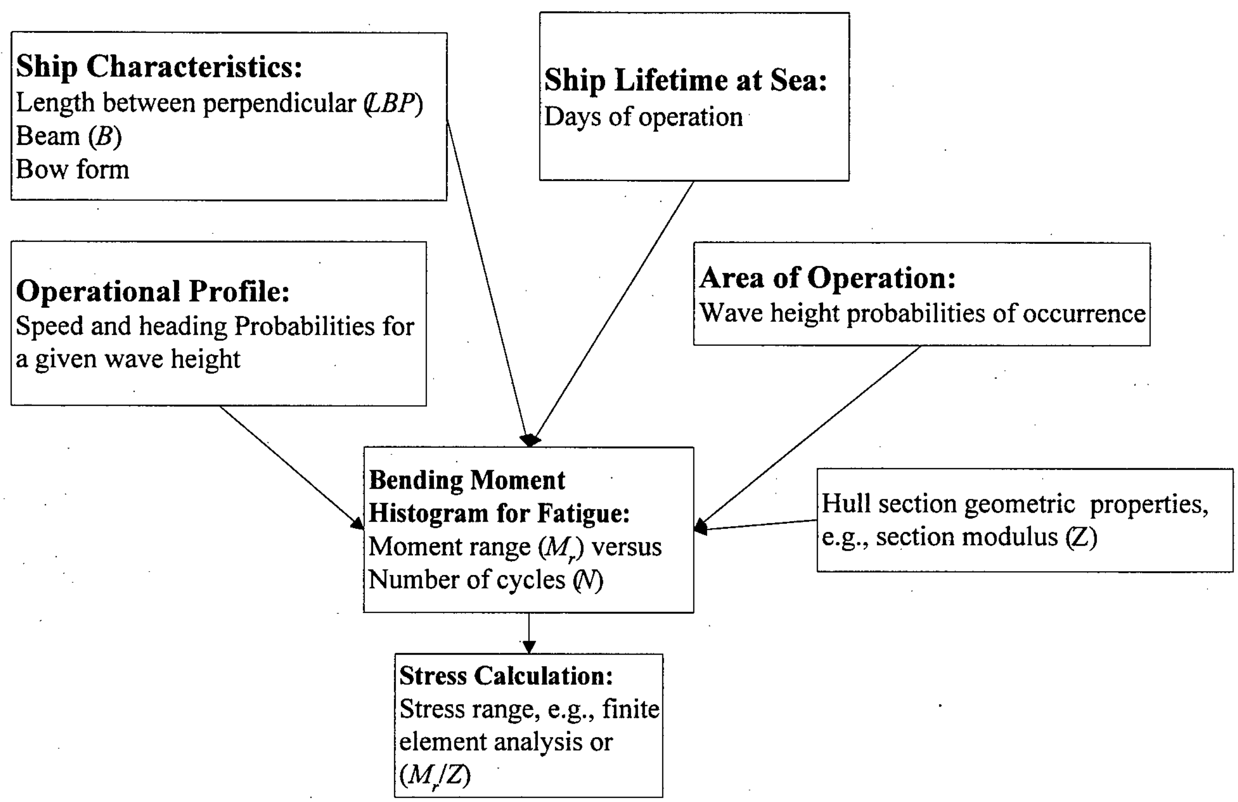

Figure 5. Response Histogram for Fatigue

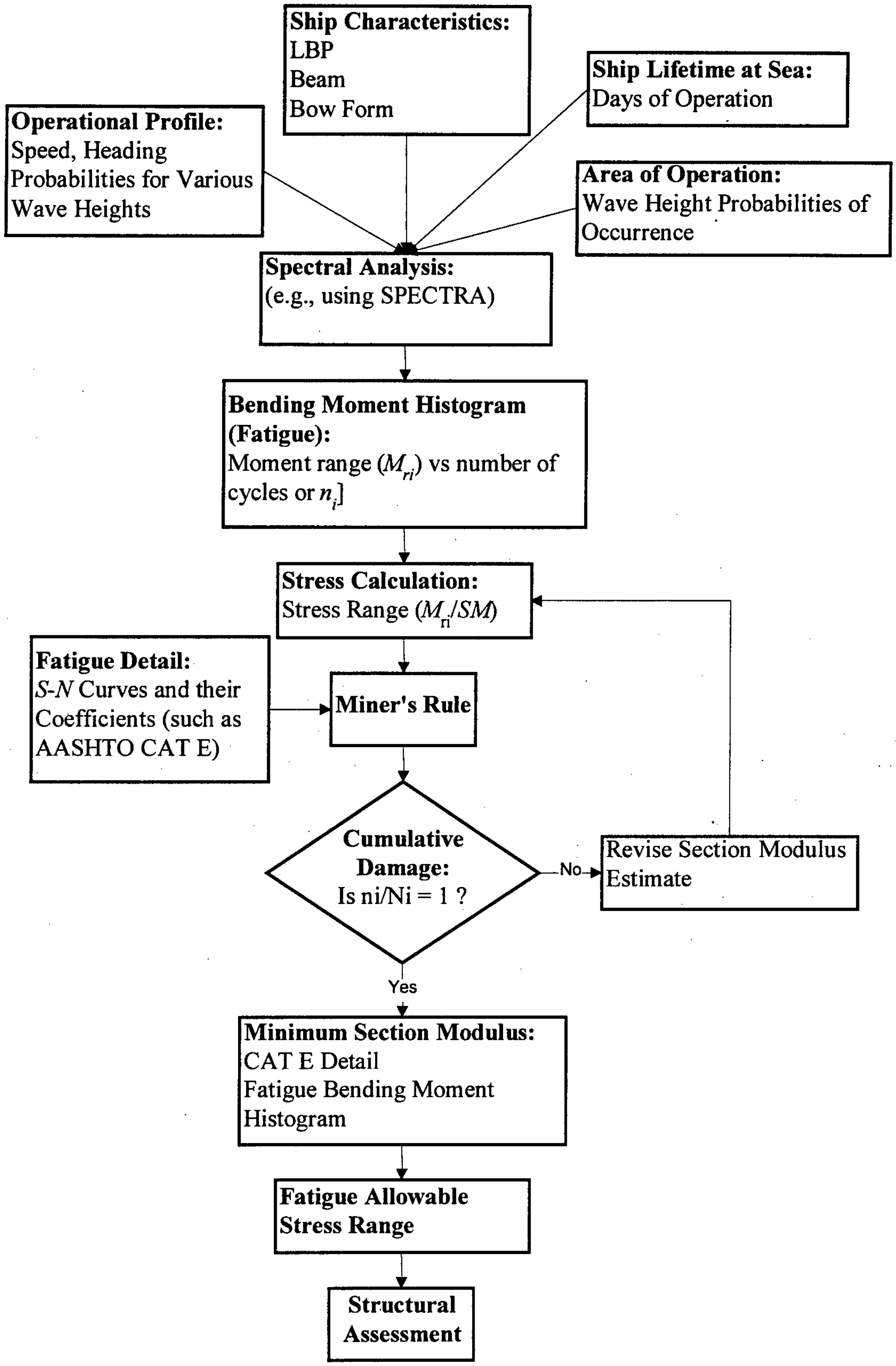

Figure 6. Design for Fatigue Based on Nominal Stress

Design Criteria

The design criteria are a service life without crack initiation. The general design procedure can be based on reliability methods. The basic fatigue methodology is shown in Figure 7. A reliability-based design requires knowledge of the service life stress histogram, which may be determined from spectral analyses of the loads. The spectral analysis shall be used to develop lifetime fatigue loads spectra by considering the operational conditions and the characteristics of a ship in the sea. The operational conditions are divided into different operation modes according to the combinations of ship speeds, ship headings, and wave heights.

The ship characteristics include the length between perpendicular (*LBP*), beam (*B*), and the bow form as shown in Figure 7. With the proper identification of the hull girder section modulus (*Z*), the bending moment histograms (moment range versus number of cycles) shall be converted to mean stress range spectra to compute the equivalent stress range according to

$$S_e = \sqrt[b]{\sum_{i=1}^{k} f_i S_i^b} \tag{15}$$

where b = slope of the *S-N* curve, S_i = stress in the i^{th} block, f_i = fraction of cycles in the ith block, and k = number of stress blocks in a stress (loading) histogram.

The equivalent stress range is sometimes useful for comparing stress histograms for fatigue crack initiation using Miner's Rule or when performing damage assessments using the fracture mechanics approach.

The reliability-based design for fatigue requires the probabilistic characteristics of the random variables in the performance function equation. It also requires specifying a target reliability index, β_0, to be compared with a computed β resulting from reliability assessment methods such as first-order reliability method (FORM) as provided by Ayyub and McCuen (Ayyub and McCuen 1997). The general form for reliability checking used in the rules is given by

$$\beta \geq \beta_0 \tag{16}$$

The performance function for fatigue can be expressed in terms of the fatigue damage ratio as follows:

$$g = \sum_{i=1}^{k} \frac{n_i}{N_i} - \Delta_L \tag{17}$$

where

Δ_L = fatigue damage ratio limit that has a mean value of one

n_i = number of actual load cycles at the ith stress-range level

N_i = number of load cycles to failure at the ith stress-range level

k = number of stress-range levels in a stress range histogram

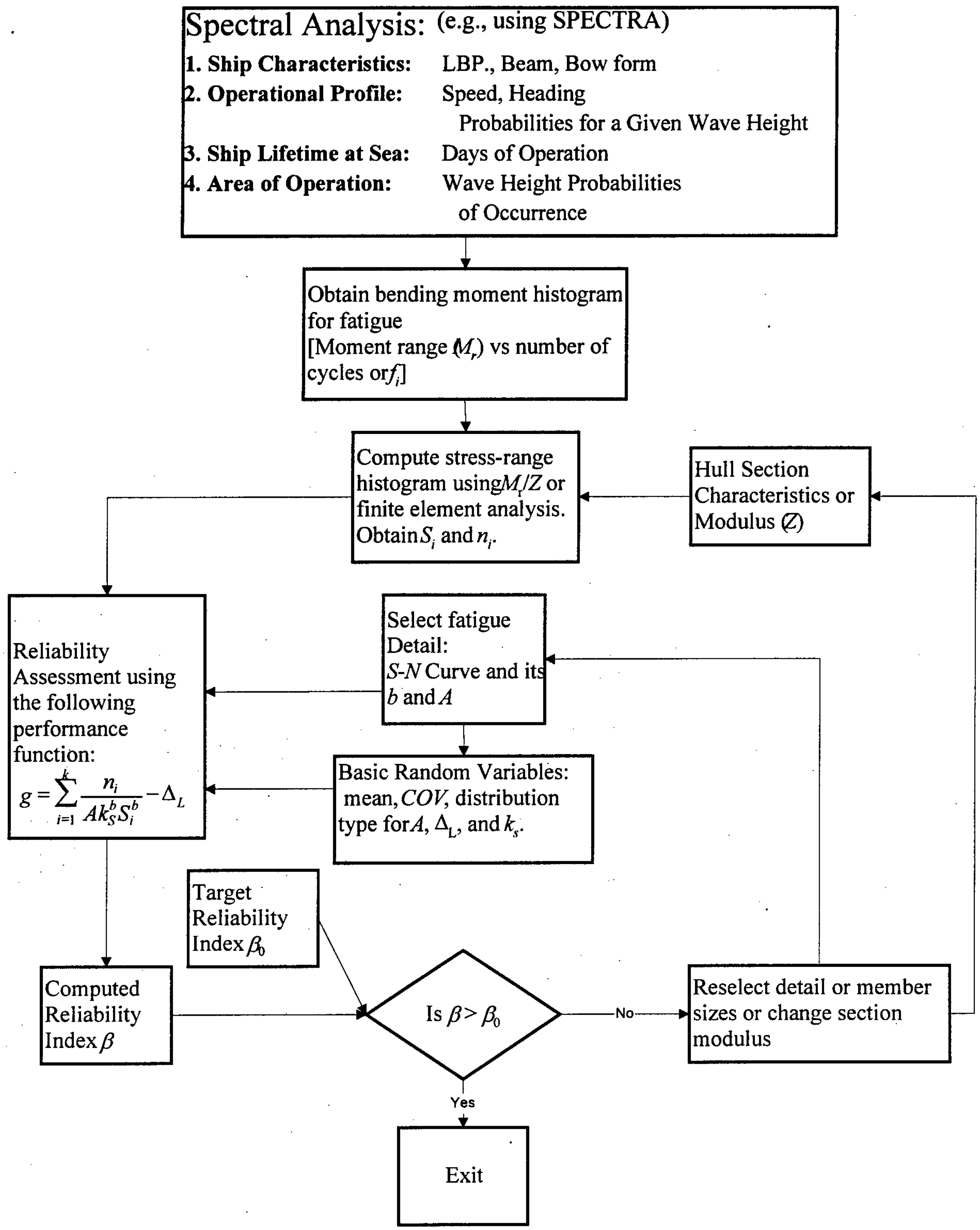

Figure 7. Reliability-based Design and Analysis for Fatigue

Using Equation 11, Equation 17 can be expressed as

$$g = \sum_{i=1}^{k} \frac{n_i}{A k_S^b S_i^b} - \Delta_L \tag{18}$$

Using two-standard deviations (2σ) from a mean regression line that represents the S-N strength of a fatigue detail, Equation 18 can be expressed as

$$g = \sum_{i=1}^{k} \frac{n_i}{10^{(\log(A)-2\sigma)} k_S^b S_i^b} - \Delta_L \tag{19}$$

where A, b and σ are obtained from linear regression analysis of S-N data in a log-log space. Equation 18 or Equation 19 can be used to perform reliability analysis and safety checking per Equation 16 as provided in Figure 7.

Cursory Design Procedure

The cursory procedure is based on results of a benchmarking effort that involved back-calculating the fatigue strength of proven ship designs. A proven ship design was considered to be one in which the ship had successfully completed its design service life without experiencing fatigue cracks. This approach assumes the benchmarked ship is just about to experience crack initiation at the end of its service life. The types of ships included three from each of the following four categories: frigates, cruisers and destroyers, auxiliaries, amphibious assault ships, and aircraft carriers. Regardless of how the ships were used during their actual lifetime, each ship was analyzed as though it operated solely in the North Atlantic to establish a design scenario. An operational profile was constructed for each type of ship based on ship operational data. Fatigue analyses were performed on each ship, parametrically varying only the number of days per year of the ship's service life at sea, for design simplicity. The stress range, identified as the permissible stress level, was determined from the maximum lifetime wave-induced plus whipping bending moment, minimum section modulus and structural detail category necessary to attain a fatigue life equal to the design service life. Actual stress levels along the length of the ship, using the same maximum lifetime wave-induced plus whipping moment, would then be compared to the permissible stress level. The optimal permissible stress level either met or slightly exceeded all actual calculated stress levels along the length of each ship within a given type. It is recognized that the ship will not spend its entire life operating the North Atlantic. The resulting total number of days in the North Atlantic is intended to represent approximately 35% operability in a less severe, but more typical, operating area.

The fatigue calculations were performed using Miner's cumulative damage hypothesis, stress range, and S/N curves associated with structural detail categories listed in the American

Association of State Highway and Transportation Officials (AASHTO) bridge fatigue design guidance (AASHTO 1992). The AASHTO curves were chosen because they are based on large-scale tests of welded steel members (NCHRP 1986) and felt to be applicable to welded ship structure. The AASHTO curves also represent a 2.3% probability of failure, providing a lower bound for design. Section moduli for each ship were calculated from scantlings taken from ship drawings. Adjustments were made for increased plating thickness due to mill tolerance (5% increase) and effectiveness of superstructure, if present. Lifetime wave-induced plus whipping bending moments were distributed along the length of a ship according to a "1 - cosine" function.

The maximum lifetime wave-induced plus whipping bending moments obtained from the analyses were fit to functions of principal ship dimension for each ship type. In this way, lifetime bending moments, associated with similar ships, could be estimated easily without using a spectral analysis.

Given the overall dimensions and type of ship, a maximum permissible design stress is therefore provided for general categories of structural details. The maximum lifetime load magnitude and longitudinal distribution are estimated with simple algorithms, and the minimum "fatigue based" section modulus thus defined. Although the ships addressed include frigates, cruisers, destroyers, auxiliaries, amphibious assault, and aircraft carriers; they are all monohulls of conventional shape with orthogonally stiffened plate construction using ordinary and high-strength steels. Since the cursory procedure is based on ships possessing these conventional characteristics, ships that deviate from those considered cannot be designed against fatigue using this procedure, as it may not necessarily result in an acceptable design. For such cases, a designer will need to resort to the general procedure and make some assumptions in order to design the ship against fatigue. In addition, if the ship configuration or construction materials are radically different than those considered herein, some model and fatigue tests may be required to quantify the loads and fatigue strength behavior, if such information is not available in the open literature. If a ship being designed is similar to any of those ships that served as the basis of this procedure, then the resulting hull is expected to have adequate strength against fatigue cracking during its service life. In general, the design allowable stresses and design conditions stated in this procedure are expected to result in a ship that has a 2.3% probability of failure during its service life under a loading condition as defined by the underlying stress-range histogram. The allowable stresses were developed according to the following requirements: (1) spectral analysis of wave loads, (2) building on conventional codes, (3) nominal strength and load values, and (4) achieving implicit reliability levels. They are applicable to frigates, cruisers and destroyers, auxiliaries, amphibious assault, and aircraft carriers for typical 30 to 50 year service lives.

Lifetime Bending Moments Amidships

Designing for fatigue requires the computation of stresses. The calculated stress range, due to the maximum lifetime bending moment range, must be less than or equal to the permissible stress range as defined by strength of fatigue details and fatigue analyses of retired naval ships. The maximum hog and sag bending moment are therefore needed for fatigue design. For amidships (i.e., at station 10), the lifetime wave-induced plus whipping bending moments can be computed using the coefficients in Table 1 and the following:

$$BM_{\max}\ [\text{ft - ltons}] = C1\left(L^{2.5}B\right)^{C2} \tag{20}$$

The maximum bending moment range is the sum of BM_{max} (hog) and BM_{max} (sag).

$$BMR_{\max}\ [\text{ft - ltons}] = BM_{\max}(hog) + BM_{\max}(sag) \tag{21}$$

For ships with reverse tumblehome, the beam (*B*) shall be taken as the average of the maximum beam and the beam at the waterline.

Table 1. Maximum Lifetime Wave-induced plus Whipping Bending Moment Coefficients

Ship Type	Frigates, Cruisers, and Destroyers	Auxiliaries and Slow Cargo Vessels	Amphibians and Fast Cargo Vessels	Carriers
Normal Ship Life	30 year life 9 dys/yr N.Atl	30 year life 23 dys/yr N.Atl	30 year life 14 dys/yr N.Atl	40 year life 25 dys/yr N.Atl
Hog: *C1*	3.217E-04	3.504E-04	3.044E-04	2.511E-03
C2	1.038E+00	1.039E+00	1.047E+00	9.506E-01
Sag: *C1*	8.979E-05	7.878E-04	6.623E-04	6.197E-03
C2	1.113E+00	1.013E+00	1.022E+00	9.245E-01
Range: *C1*	3.218E-04	1.100E-03	9.387E-04	8.373E-03
C2	1.078E+00	1.024E+00	1.033E+00	9.355E-01
Extended Ship Life	40 year life 9 dys/yr N.Atl	40 year life 23 dys/yr N.Atl	40 year life 14 dys/yr N.Atl	50 year life 25 dys/yr N.Atl
Hog: *C1*	4.476E-04	3.710E-04	3.147E-04	2.694E-03
C2	1.022E+00	1.037E+00	1.046E+00	9.479E-01
Sag: *C1*	1.261E-04	8.033E-04	6.806E-04	6.762E-03
C2	1.096E+00	1.013E+00	1.022E+00	9.210E-01
Range: *C1*	4.500E-04	1.140E-03	9.670E-04	9.071E-03
C2	1.062E+00	1.023E+00	1.032E+00	9.323E-01

Longitudinal Distribution of Bending Moments

The magnitude of the maximum bending moment range is largest amidships and is reduced in magnitude toward the ends of the ship. The maximum bending moment range for any location is determined by multiplying the midship value by a distribution factor. This distribution factor (*DF*) for station *i* is represented by

$$DF_i = 0.5\left(1 - \cos\left(2\pi x_i / L\right)\right) \tag{22}$$

where x_i = distance from the forward perpendicular to the section of interest. The *DF* for all 20 stations is given in Table 2 and Figure 8.

The maximum bending moment range at station *i* is represented longitudinally by

$$BMR_{\max i} = DF_i\left(BMR_{\max}\right) \tag{23}$$

Table 2. Longitudinal Distribution of Maximum Bending Moment Range

Station	Distribution factor (DF)
0	0.00000
1	0.02447
2	0.09549
3	0.20611
4	0.34549
5	0.50000
6	0.65451
7	0.79389
8	0.90451
9	0.97553
10	1.00000
11	0.97553
12	0.90451
13	0.79389
14	0.65451
15	0.50000
16	0.34549
17	0.20611
18	0.09549
19	0.02447
20	0.00000

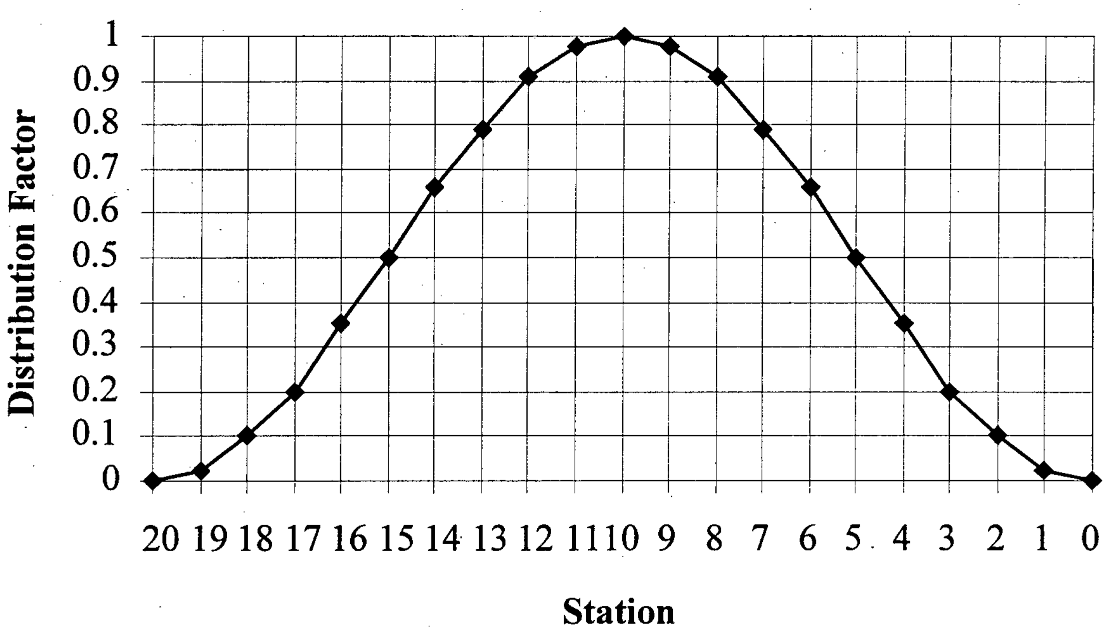

Figure 8. Longitudinal Distribution Factor (*DF*)

The speed and heading probabilities used to develop the cursory design guidelines are provided in Table 3 through Table 6. Wave height probabilities and ship characteristics defined in Table 7 and Table 8 were used to estimate the lifetime bending moments for the different types of ships. The Ochi six-parameter family of spectra was used to generate lifetime fatigue spectra by computing the number of cycles for a specified response. The number of days per year spent in the North Atlantic is the time that was necessary to accumulate unit fatigue damage; that is, fatigue life equal to service life. This criterion was established for design purposes as a severe condition. This severe design condition is intended to represent approximately 35% operability in a less severe, but more typical area.

Table 3. Speed and Heading Probabilities for Frigates, Cruisers and Destroyers

(a) Low Sea States (0-3 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.06884	0.10590	0.01583
Bow	0.09032	0.15724	0.02168
Beam	0.06641	0.10414	0.01473
Stern Qtr	0.07458	0.13127	0.01815
Follow	0.04791	0.07297	0.01004

(b) Medium Sea States (3-6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.06131	0.14975	0.01809
Bow	0.09045	0.19698	0.02714
Beam	0.04422	0.1397	0.01307
Stern Qtr	0.04422	0.09447	0.02412
Follow	0.02714	0.061314	0.00804

(c) High/extreme Sea States (>6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.11111	0.16049	0
Bow	0.08642	0.16049	0.01235
Beam	0.08642	0.09877	0.02469
Stern Qtr	0.02469	0.08642	0.01235
Follow	0.02469	0.04938	0.06173

Table 4. Speed and Heading Probabilities for Auxiliaries, Tankers and Slow Cargo Vessels

(a) Low Sea States (0-3 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.04278	0.15471	0
Bow	0.06082	0.20694	0
Beam	0.04243	0.14620	0
Stern Qtr	0.04486	0.15957	0
Follow	0.03393	0.10777	0

(b) Medium Sea States (3-6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.05491	0.19042	0
Bow	0.04439	0.26636	0
Beam	0.03271	0.13318	0
Stern Qtr	0.03271	0.14836	0
Follow	0.02220	0.07477	0

(c) High/extreme Sea States (>6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.04938	0.19753	0
Bow	0.11111	0.19753	0
Beam	0.02469	0.18519	0
Stern Qtr	0.03704	0.14815	0
Follow	0	0.04938	0

Table 5. Speed and Heading Probabilities for Amphibious Assault and Fast Cargo Vessels

(a) Low Sea States (0-3 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.06345	0.11891	0.00174
Bow	0.10554	0.17500	0.00190
Beam	0.07840	0.10585	0.00190
Stern Qtr	0.08544	0.13125	0.00269
Follow	0.04984	0.07650	0.00158

(b) Medium Sea States (3-6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.06341	0.16304	0.01268
Bow	0.08696	0.27899	0.01087
Beam	0.03261	0.12681	0
Stern Qtr	0.03442	0.12681	0
Follow	0.02174	0.03804	0.00362

(c) High/extreme Sea States (>6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.05714	0.22857	0
Bow	0.08571	0.25714	0
Beam	0.05714	0.05714	0
Stern Qtr	0.05714	0.11429	0
Follow	0.02857	0.05714	0

Table 6. Speed and Heading Probabilities for Aircraft Carriers

(a) Low Sea States (0-3 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.07428	0.10083	0.02285
Bow	0.11625	0.09991	0.03721
Beam	0.08851	0.08973	0.02728
Stern Qtr	0.09484	0.06383	0.03055
Follow	0.05235	0.04625	0.01860

(b) Medium Sea States (3-6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.10083	0.06846	0.01943
Bow	0.09991	0.15171	0.03608
Beam	0.08973	0.11193	0.04163
Stern Qtr	0.06383	0.07031	0.02405
Follow	0.04625	0.06105	0.01480

(c) High/extreme Sea States (>6 meters)

Heading	Speed (knots)		
	5	15	25
Head	0.03846	0.03846	0
Bow	0.30769	0.11538	0
Beam	0.11538	0.15385	0.07692
Stern Qtr	0.07692	0.03846	0.03846
Follow	0	0	0

Table 7. Ship Characteristics and Operational Baseline Parameters

Characteristics and Parameters	Frigates, Cruisers, and Destroyers	Auxiliaries, Tankers and Slow Cargo Vessels	Amphibious Assault and Fast Cargo Vessels	Aircraft Carriers
Bow Form	Fine	Flat-Bottomed	Flat-Bottomed	Large Flair
Displacement	< 10,000 ltons	> 15,000 ltons	>10,000 ltons	>75,000 ltons
Speed and Heading Probabilities	Table 3	Table 4	Table 5	Table 6
Days Operation in North Atlantic (per year)	9	23	14	25

Table 8. Wave Height Probabilities

Significant Wave Height (meters)	% Probability of Occurrence
0-1	10.14
1-2	20.31
2-3	20.35
3-4	16.04
4-5	12.14
5-6	8.00
6-7	4.85
7-8	3.39
8-9	2.09
9-10	1.16
10-11	0.68
11-12	0.40
12-13	0.21
13-14	0.16
14-15	0.10
15-16	0.00

Stress Analysis

Depending on the complexity of the structure, this step can either be accomplished with simple strength of materials calculations, or a finite element model. The objective here is to establish the relationship between the applied loads, in this case, longitudinal vertical bending moment, and applied stress at the point of interest within the hull girder. Since applied stress associated with fatigue *S-N* data is generally nominal far field, using strength of material calculation or a coarse mesh finite element model is generally sufficient. However, global stress concentrations due to deck openings, misalignments, and other structural discontinuities must be accounted for and included when computing the applied stress. Depending on the database used to generate the *S-N* curves, local stress concentrations due to weld and specimen geometry are generally ignored since they are usually integral to the specimen when fabricated. Effects of secondary loads or lateral bending on fatigue damage accumulation are not explicitly addressed in the cursory procedure.

Once the maximum lifetime applied load range is known, the permissible design stress range can be used to determine the minimum section modulus for a given category detail. Categories associated with the AASHTO design *S-N* curves can be used. Generally, for typical ship design, a Category E detail will best represent the intersecting details of orthogonally stiffened plating. Permissible stresses associated with other categories are provided for structural situations that are either more severe or less severe than typical ship structure. Fatigue analyses do not need to be performed. It is assumed that the fatigue analyses already performed on retired naval ships will lead to an acceptable design against failure by fatigue. Although a probabilistic design was not performed, the permissible stress ranges are associated with a 2.3% probability of exceedence based on only the scatter of the *S-N* data.

Hull-Girder Section and Strength Properties

The cross-sectional properties of hull girders shall be defined by using the inertia sections developed in the Longitudinal Strength Drawing (DDS 100-6) (Bureau of Ships 1950). Inertia sections are required for stations 1 through 19. The moment of inertia curve shall be free of sharp rises and depressions. Structural components that do not meet buckling or tripping criteria should not be included in these calculations.

Longitudinal Stress Distribution

The nominal longitudinal stress-range distribution, for stations of interest, shall be calculated by dividing the longitudinal bending moment range for each station, determined in the Lifetime Bending Moments Amidships Section, page 21 and the Longitudinal Distribution of Bending Moments Section, page 22 of these guidelines, by the section modulus for that station. Stresses between stations can be considered to vary linearly, and hence linear interpolation can be used for this purpose.

For internal longitudinal strength structure on centerline, the stresses shall be assumed to taper linearly to zero at the hull girder neutral axis; for shell structure the stress range shall taper to one-half the extreme fiber stress range. For internal longitudinal structure off centerline, the stress shall taper from the shell stress to the centerline stress. This accounts for lateral bending components that are not explicitly considered in this procedure.

Stress Concentrations

Stress concentrations increase the stress range at the detail above the nominal stress level. All weld toes, transitions and attachments represent points of localized stress concentration. A local stress concentration need not be considered because its effects are inherent in the *S-N* curve for the joint. However, if the weld detail itself lies in an area of gross stress concentration due to the geometry of the ship structure, this gross stress concentration and associated peak stress must be quantified and accounted for in the calculations.

An analytical method for calculating stress concentration factors (SCFs) for openings and plate misalignment is provided in Appendices A and B, respectively. For closely spaced openings and any other configurations beyond the applicability of Appendices A and B, SCFs shall be computed using accepted practices.

Attaching the superstructure to the strength deck requires special consideration at the fore and aft ends. Finite element method (FEM) analysis should be used as described in the General Design Procedure, page 6.

Peak Stress Range

In areas of stress concentration, peak stress range is critical to fatigue life. A peak stress range is defined as the product of the nominal stress range and an appropriate stress concentration factor. So long as the calculated peak stress range at any point is less than the permissible stress range, an acceptable design against fatigue is achieved.

Fatigue Damage Accumulation

The basis of the permissible stress ranges is made through the use of Miner's linear cumulative damage hypothesis. This hypothesis states that the fatigue damage caused by any stress cycle is independent of neighboring stress cycles; there is no retardation or acceleration of micro-crack coalescence. Furthermore, the fatigue damage is assumed to accumulate as the ratio of the applied number of stress cycles to the number of stress cycles to cause failure.

S-N Curves

The fatigue strength of a structural detail is defined by a curve of stress range versus number of cycles to failure (*S-N* curve). Fatigue strength varies from detail to detail. AASHTO has grouped details into categories A to E' in order of increasing severity (see Representative Details, below). Each category has a representative *S-N* curve illustrated in Figure 9. Each *S-N* curve is defined by coefficients, log(A) and b in Table 9, based on a mean minus two standard deviation regression line as a safety measure (Fisher and others 1993).

The curves in Figure 2 are applicable to steels with yield strength up to 80 ksi. The curves represent in-air performance that is applicable because of the Navy's coating and cathodic protection system.

The number of cycles to failure (N) of a detail at a stress range (S) is calculated using the following equation that can be obtained from Equation 12:

$$N = 10^{[\log A + b \log(S)]} \tag{24}$$

The AASHTO curves were used to benchmark the procedure documented here by back calculating the fatigue strength of retired ships, which have experienced no cracking. As such, use of the curves is conservative and would limit the risk of crack initiation as represented by 2.3% probability of crack initiation at any given constant amplitude stress range.

Table 9. *S-N* Curve Coefficients (Stress Ranges in ksi)

Fatigue Detail Category	*B*	Log(*A*)
A	-3.0	10.401
B	-3.0	10.080
B'	-3.0	9.791
C	-3.0	9.652
D	-3.0	9.335
E	-3.0	9.030
E'	-3.0	8.583

Representative Details

AASHTO has provided a grouping of structural details based on fatigue strength. This grouping, ranging from category A to category E', depends on the severity of the detail and its susceptibility to crack initiation. Details are provided in Table 10, Figure 9, Appendix C, Fisher (1993), and AASHTO (1992). Ranking of details assumes fabrication performed as provided in MIL-STD-1689 (Department of Defense 1983).

The detail most critical to the fatigue life of a strength deck and shell near the keel, in terms of nominal stress, is the intersection of a transverse bulkhead stiffener and deck or shell longitudinal. This detail is category E and prevalent in primary hull structure.

The detail applicable to a full-penetration transverse butt weld is category C. The rat hole in a deck or shell longitudinal is a category D detail if its diameter is less than 4 inches and category E if it's diameter is 4 to 6 inches.

Thick plates are often inserted into structure for reinforcement or ballistic protection. The applicable detail is dependent on the chamfer of the thicker plate. If the thick plate chamfer is greater than or equal to 2.5:1, then the applicable detail is category C. If the thick plate chamfer is (steeper) less than 2.5:1, then the applicable detail is category D.

Openings in primary hull structure are common. The detail applicable to the peak stress at an opening is dependent on the method of reinforcement. A non-reinforced opening or an opening reinforced with an insert plate has a flame cut edge at the location of peak stress. A flame cut edge is a category C detail.

An opening reinforced with a ring has a longitudinal fillet weld at the area of peak stress. A longitudinal fillet weld is a category B detail. Complete penetration longitudinal welds with permanent backing bars, as well as partial penetration longitudinal welds, are a category B' details. A reinforcing ring is often fabricated from several pieces of flat bar or plate which are butt welded with full penetration welds. The full penetration transverse butt weld detail category C is applicable to the ring butt.

Chocks or brackets are often attached to the coamings and deck by a fillet weld. These attachments are category C details. If the coaming butt weld or attachment is located close to the area of peak stress of the opening, these details are more critical than the longitudinal fillet welds.

A welded knuckle has a full penetration transverse butt weld and rat-hole in the longitudinal stiffener at the location of peak stress.

Table 10. Design Details and Category

Detail	Category
Base metal, rolled shapes, machined ground flame cut edges with ANSI /ASME (ANSI/ASME 1985) smoothness of 1000 micro-inches or less	A
Continuous longitudinal fillet weld	B
Full penetration longitudinal fillet weld with permanent backing bar and partial penetration longitudinal welds (possible corrosion effects excluded)	B'
Transverse butt joint	C
Transverse butt joint with plates of unequal thickness and:	
Transition greater than or equal to 2.5:1	C
Transition less than (steeper) than 2.5:1	D
Non load carry attachment shorter than 2"	C
Cruciform joint where loaded member continuous	C
Flame cut edge	C
Non-load carrying attachment between 2" and 4" long	D
Transverse frame or floor at shell or deck	D
Rat hole < 4" long	D
Non-load carrying attachment longer than 4" and < 1" thick	E
Load carrying attachment < 1" thick	E
Rat hole > 4" long	E
Non-load carrying attachment longer than 4" and > 1" thick	E'
Load carrying attachment > 1" thick	E'

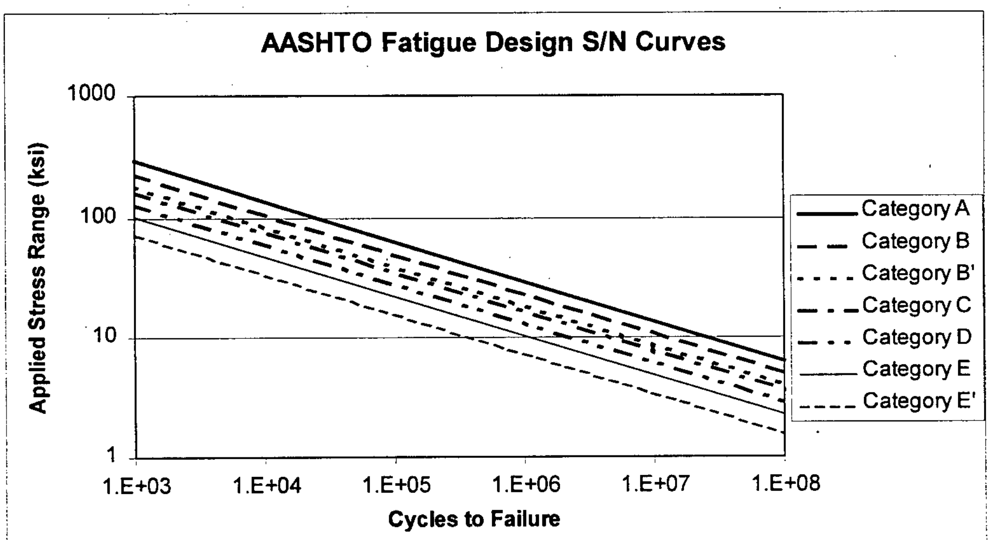

Figure 9. AASHTO Fatigue Design *S-N* Curves

Design Criteria

Fatigue design criteria for surface ships involve first limiting the nominal stress range in the structure based on a critical detail. The structure and its details are then designed to a criteria intended to minimize stress concentrations.

Nominal Stresses

A calculated nominal stress range, due to the maximum bending moment range, shall be less than or equal to the permissible stress range for a category E detail. The permissible stress ranges for service lives of 30 and 40 years for combatants, auxiliaries and amphibious assault ships and 40 and 50 years for aircraft carriers are provided in Table 11. For the required ship service life, governing specifications shall be used. The calculated stress range shall not exceed twice the tensile yield strength of the material.

In those areas where details more severe than CAT E are unavoidable, local structural reinforcement shall be provided and the calculated stress range at the detail shall be less than or equal to the permissible stress range for the detail.

Table 11. Permissible Stress Ranges (ksi)

(Not to exceed twice the tensile yield strength of the material)

Category	Frigates, Cruisers, and Destroyers	Auxiliaries and Slow Cargo Vessels	Amphibians and Fast Cargo Vessels	Aircraft Carriers
Normal Ship Life	30 years	30 years	30 years	40 years
A	141	116	132	127
B	110	91	103	99
B'	88	73	83	79
C	80	65	74	71
D	62	51	58	56
E	49	41	46	44
E'	35	29	33	31
Extended Ship Life	40 years	40 years	40 years	50 years
A	131	107	122	119
B	102	84	95	93
B'	82	67	76	75
C	74	60	68	67
D	58	47	54	53
E	46	37	42	42
E'	32	27	30	29

Permissible Stress Concentration Factors

Stress concentration factors in way of openings subject to primary bending stresses shall not exceed the maximum permissible SCF's for the appropriate detail. Openings in the deck stringer strakes and shell sheer strakes require consideration of the complex stress field resulting from primary bending and shear, and superstructure uplift force.

Table 12 shows the maximum permissible SCF for opening as a function of detail category. The permissible SCF for a category of detail is the permissible stress range for that detail divided by the permissible stress range for the critical Category E detail. Stress concentration factors for internal structure at the neutral axis are derived by doubling the stress concentration factors for the hull girder envelope. This increase in SCF for internal structure is applied to compensate for the current design assumption that the vertical stress distribution for the outer hull envelope does not taper to zero at the neutral axis, but rather to half the value at the extreme fiber.

Table 12. Maximum Permissible Stress Concentration Factor

Detail Category	Maximum Permissible Stress Concentration Factor Note 1	
	Hull Girder Envelope Note 2	Interior Structure at the Neutral Axis
A	2.8	5.6
B	2.2	4.4
B'	1.8	3.6
C	1.6	3.2
D	1.2	2.4
E	1.0	2.0
E'	0.7	1.4

Notes:

1. The maximum permissible SCF shall be interpolated linearly for interior structure between those values provided for the extreme fiber of the hull girder and the neutral axis.
2. The hull girder envelope consists of the shell, inner bottom, double bottom plate longitudinals, and uppermost strength deck.

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Appendix A
Methods for Calculating Stress Concentration Factors (SCF) for Openings

Symbols and Abbreviations

B = deck width (inches)

b = opening width (inches)

a = opening length (inches)

r = opening radius (inches)

t_p = nominal plate thickness (inches)

t_i = insert plate thickness (inches)

t_c = reinforcing ring thickness (inches)

h = reinforcing ring height (inches)

D = distance from centerline of opening to centerline of ship (inches)

C = shortest distance from centerline of opening to deck edge (inches)

S_1 = shortest distance from edge of opening to deck edge (inches)

S_2 = longest distance from edge of opening to deck edge (inches)

k_{bo} = SCF for opening in infinitely wide plate

k_{b1} = SCF for eccentrically located rectangular opening in a finite width plate

k_b = SCF for opening in finite width plate

β_0 = stress reduction factor due to reinforcing ring

β_1 = stress reduction factor due to reinforcing ring

Θ = stress reduction factor due to insert plate

x = longitudinal distance from forward perpendicular (feet)

L = length of the ship in feet, defined as the distance between perpendiculars LBP

Calculating SCF for Openings

Openings disrupt the nominally uniform flow of stress in the plating structure, producing high or peak stresses around the opening. These high stresses can be evaluated in terms of stress concentration factors (SCFs).

The SCF for an opening is primarily a function of length to width aspect ratio, and the ratio of corner radius to width (geometry effects), width relative to the width of the ship deck (finite plate effect), location relative to the centerline of the ship (eccentricity effects), and whether or not insert plates and edge coamings are used (reinforcement effects). Other effects such as hole proximity, coaming height symmetry, and complex opening geometries are not considered here but should be addressed with FEM analysis where applicable.

For a circular opening, the maximum stress concentration factor for an infinitely wide plate, k_{bo}, of three occurs at the edge tangent to the direction of applied stress (Boresi and Sidebottom 1985). A k_{bo} of negative one occurs at the edge perpendicular to the direction of applied stress. The direction of stress associated with the maximum k_{bo} is in the direction of applied stress, and the direction of stress associated with a k_{bo} of negative one is in a direction perpendicular to the applied stress.

For square and rectangular openings the maximum k_{bo} occurs near the corner; the stress of which is in a direction tangent to the opening edge (Brock 1957; Sobey 1962; Sobey 1963). A minimum k_{bo} of negative one exists at the midpoint of the edge perpendicular to the direction of applied stress. This is the optimum location for a coaming butt weld.

SCF for Circular Openings

For circular openings that are eccentrically (or centrally) located in a finite width plate, the SCF, k_b, is given by Roark (1975).

$$k_b = \left[3.00 - 3.13\frac{r}{C} + 3.66\left(\frac{r}{C}\right)^2 - 1.53\left(\frac{r}{C}\right)^3\right]F \tag{A-1}$$

where

$$F = \frac{\left(1-\frac{C}{B}\right)\left[1-\left(\frac{r}{C}\right)^2\right]^{0.5}}{\left(1-\frac{r}{C}\right)\left[1-\frac{C}{B}\left(2-\left(1-\left(\frac{r}{C}\right)^2\right)^{0.5}\right)\right]} \tag{A-2}$$

Figure A-1 shows the variables used to describe a circular opening and to determine the SCF. Figure A-2 can also be used to determine k_b. The variables r and C in Eqs. A-1 and A-2 represent the radius of the circular opening and the distance from the deck edge to the center of the opening.

SCF for Square and Rectangular Openings

For square and rectangular openings of width, b, which are eccentrically (or centrally) located a distance S_1 from the edge of a finite width plate, the *SCF* at the corners of side S_1, k_{b1}, is given by

$$k_{b1} = \frac{\alpha\left[1+\dfrac{b}{2S_2}\right]\left(1+\dfrac{b}{S_1}\right)^{0.5}}{1-\dfrac{S_1}{S_2}+\left[\left(\dfrac{S_1}{S_2}\right)^2+\left(\dfrac{S_1}{S_2}\right)\left(\dfrac{b}{S_2}\right)\right]^{0.5}} \tag{A-3}$$

where

$$\alpha = k_{bo}\left[\frac{1}{1+\dfrac{b}{2S_1}}\right]+\left[\frac{1}{1+\dfrac{2S_1}{b}}\right] \tag{A-4}$$

Figure A-3 shows the variables used to describe a square opening and determine the SCF. The parameter k_{bo} in Equation A-4 is the *SCF* for an infinite plate which can be determined from Figure A-4 (Heller Jr., Brock, and Bary 1959). For SCFs on side S_2, Equation A-4 can be used by exchanging the subscripts 1 and 2. Table A-1 provides interpolation polynomials for computing k_{bo} as a function of $p=r/b$ and a/b. These polynomials were fit to computed analytical results in order to provide closed-form solutions that can be used easily in computer programs or spreadsheets for computing SCF; however, in using these equations, the value of p should be limited to the range 0.03 to 0.5.

Table A-1. Approximate Polynomials for Computing k_{bo}

a/b	Basic Stress Concentration Factor
1/4	$k_{bo} = 39113.55p^4 - 10420.7p^3 + 1393.209p^2 - 114.40p + 8.5452$
2/7	$k_{bo} = 31246p^4 - 9287.2p^3 + 1270.1p^2 - 105.24p + 8.197$
1/3	$k_{bo} = -98825p^5 + 62614p^4 - 13800p^3 + 1542.2p^2 - 105.48p + 7.8016$
2/5	$k_{bo} = 7399.3p^4 - 3008.4p^3 + 536.42p^2 - 57.034p + 6.6419$
1/2	$k_{bo} = 3666.5p^4 - 1864.3p^3 + 385.14p^2 - 45.256p + 6.0534$
2/3	$k_{bo} = 556.69p^4 - 427.67p^3 + 158.54p^2 - 32.201p + 5.65$
1	$k_{bo} = 162.5p^4 - 240.31p^3 + 132.73p^2 - 31.712p + 5.5524$
1 1/2	$k_{bo} = -490.15p^5 + 811.53p^4 - 527.14p^3 + 176.51p^2 - 32.546p + 5.2309$
2	$k_{bo} = -393.03p^5 + 587.88p^4 - 338.24p^3 + 103.84p^2 - 20.415p + 4.4096$
3	$k_{bo} = 14.817p^3 - 4.222p^2 - 4.0363p + 3.3316$
4	$k_{bo} = 17.924p^3 - 7.0013p^2 - 3.2419p + 3.1902$
ovaloid	$k_{bo} = 14.631p^2 - 14.941p + 6.8402$
Locus of Minimums	$k_{bo} = -131.59p^3 + 116.48p^2 - 38.593p + 7.3137$

Effect of Reinforcement on the SCF around Openings

A reinforcing ring and insert plate may be used individually or in combination to reduce stress concentrations. Improper use of reinforcement material can result in greater maximum stress levels due to adverse redistribution of stress.

Reinforcing Ring

The effect of the reinforcing ring on the *SCF* is illustrated in Figure A-5 for a ring thickness equal to the nominal or insert plate thickness and ring height to deck thickness ratio, *h/t*, equal to 7. The stress reduction factor, β_0, is a function of opening corner radius to width ratio, *r/b* as shown in the figure. For ring heights beyond eight times the thickness of deck plate ($h/t > 7$), no further reduction is realized. For $r/b = 0.5$, the limiting case of a circular opening, $\beta_0 = 0.55$.

For a ring thickness equal to 1.2 times the deck plating thickness, a further reduction factor, β_1, of 0.96 is employed. It can be assumed that, for a ring thickness between 1.0 and 1.2 times the plating thickness, a linear interpolation between 1.0 and 0.96, respectively, applies for β_1. For ring dimensions outside the range recommended in Figure A-5, the reduction in *SCF* must be determined using FEM.

The maximum stress concentration factor at an opening is therefore adjusted for ring reinforcement effects as shown below.

$$SCF_{\max} = k_{b1}\beta \qquad \text{(A-5)}$$

where

$$\beta = \beta_0\beta_1 \qquad \text{(A-6)}$$

Insert Plates

Effects of insert plates are also taken into account when determining the maximum stress concentration factor at openings. The reduction factor due to the presence of an insert plate is

$$\Theta = \left[1 - \left(\left(t_i - t_p\right)/\left(3t_p\right)\right)\right] \qquad \text{(A-7)}$$

where t_i = insert plate thickness, and t_p = parent plate thickness.

Maximum SCF *at Opening*

Using the reduction factors of Equation A-6 and Equation A-7, the maximum *SCF* at the opening is

$$SCF_{\max} = k_{b1}\mathrm{B}\Theta \qquad \text{(A-8)}$$

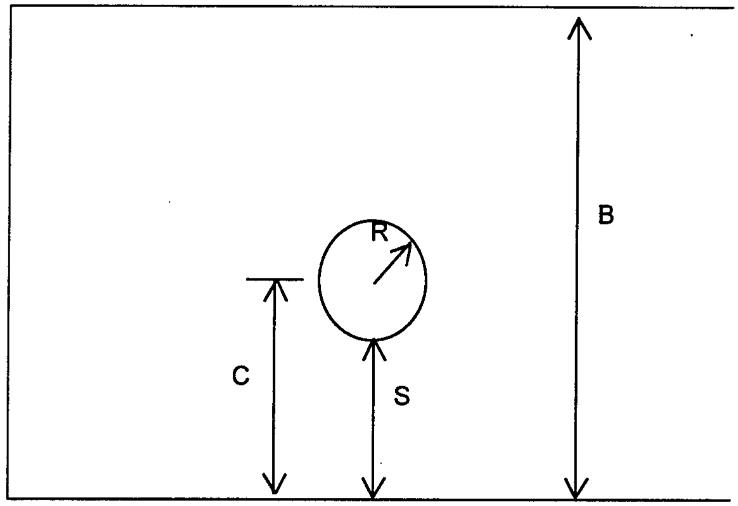


Figure A-1. Variables Used to Describe a Circular Opening

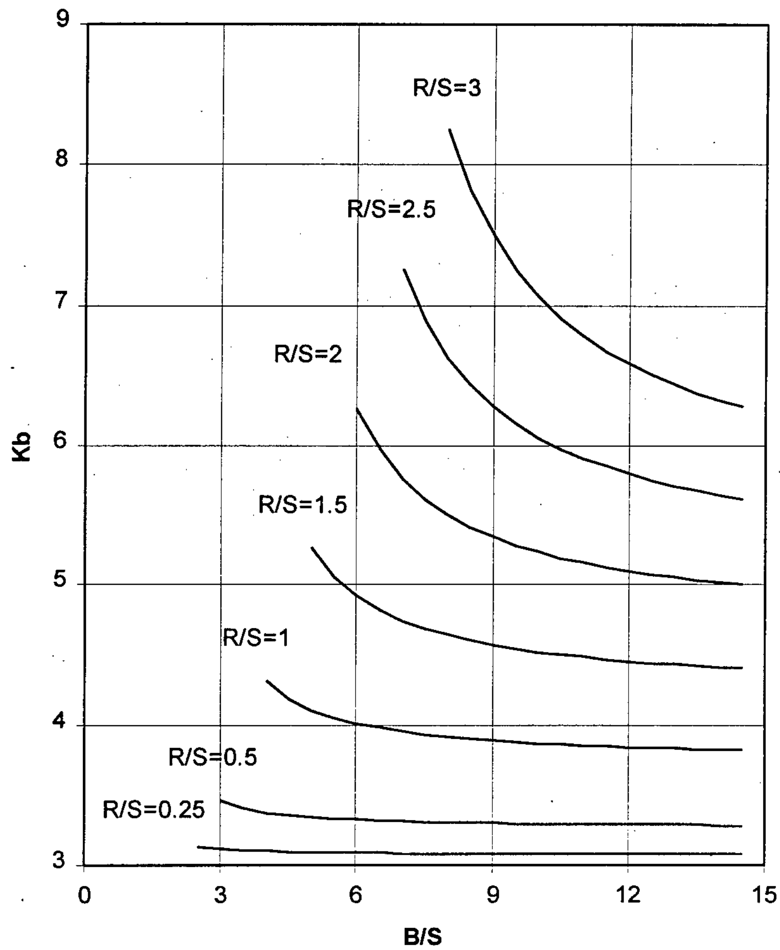


Figure A-2. Basic SCF for Circular, Eccentric Openings in a Finite Width Plate

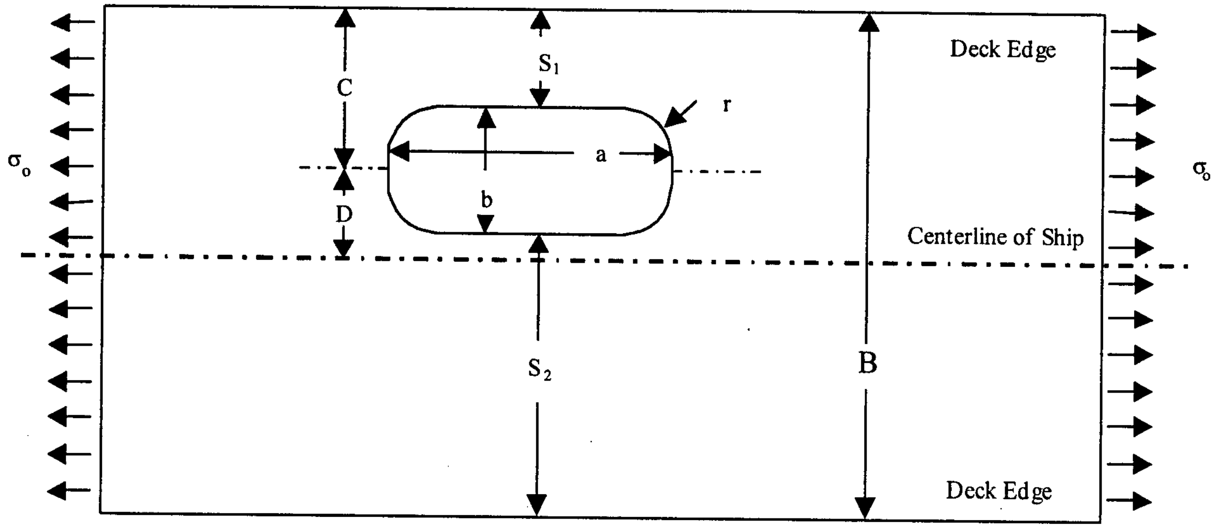


Figure A-3. Variables Used to Describe a Rectangular Opening

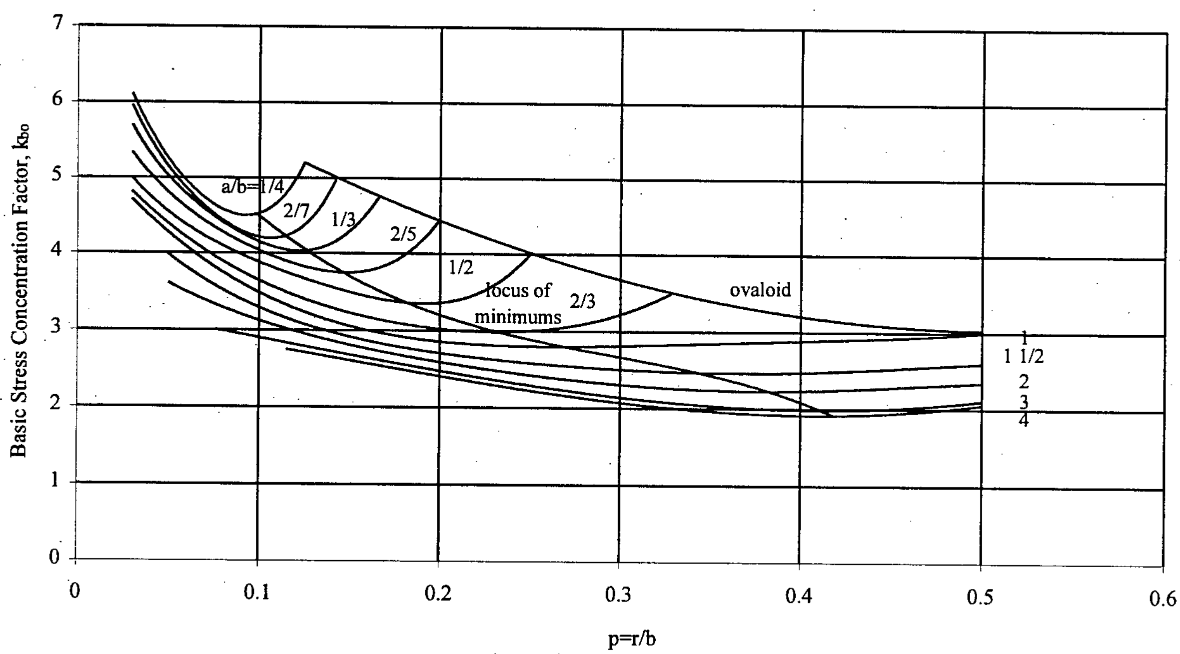


Figure A-4. Basic SCF for Square and Rectangular Opening in an Infinitely Wide Plate

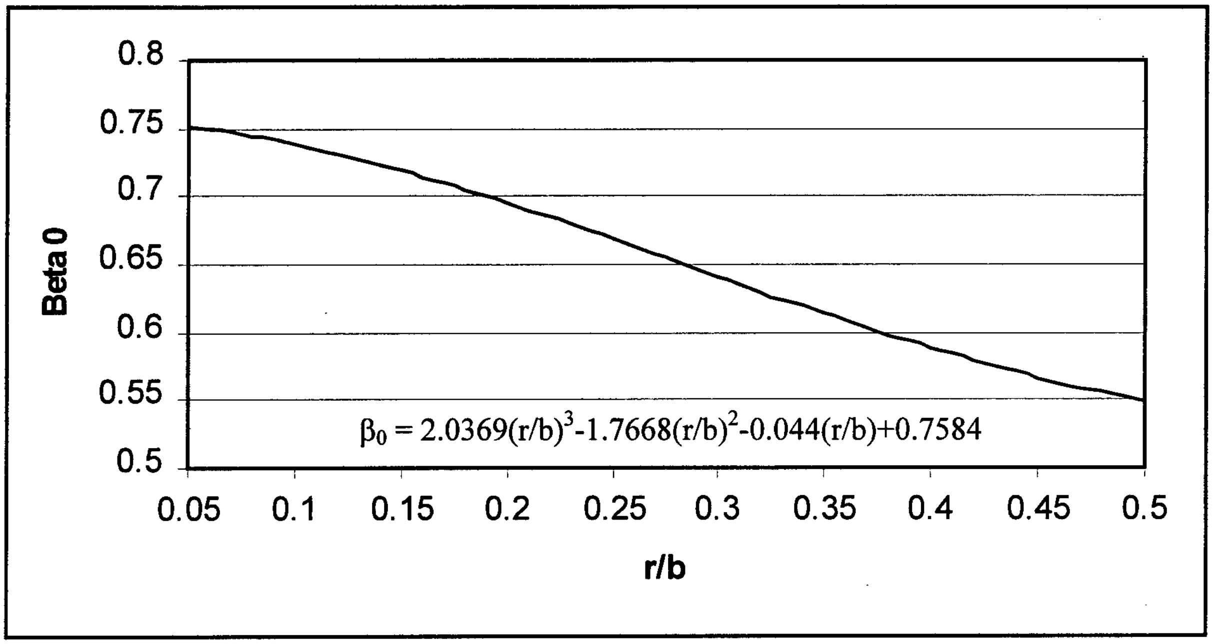


Figure A-5. Reduction in SCF due to a Symmetrical Reinforcing Ring

References

Boresi, A.P., and O.M. Sidebottom. 1985. *Advanced Mechanics of Materials*. 4th ed. New York, N.Y.: John Wilely & Sons.

Brock, J.S. 1957. Analytical Determination of the Stresses Around Square Holes with Rounded Corners, DTMB No. 1149. David Taylor Model Basin. West Bethesda, MD

Heller Jr., S.R., J.S. Brock, and R. Bary. 1959. The Stresses around a Rectangular Opening with Rounded Corners in a Uniformly Loaded Plate, DTMB No. 1290. David Taylor Model Basin. West Bethesda, MD

Roark, R.J., and W.C. Young. 1975. *Formulas for Stress and Strain*. 5th ed. New York, N.Y.: McGraw-Hill Book Company.

Sobey, A.J. 1962. The Estimation of Stresses around Unreinforced Holes in Infinite Elastic Sheets, Reports and Memoranda No. 3354. Aeronautical Research Council.

Sobey, A.J. 1963. Stress Concentration Factors for Rounded Rectangular Holes in Infinite Sheets, Reports and Memoranda No. 3407. Aeronautical Research Council.

Appendix B
Methods for Calculating Stress Concentration Factors (SCF) due to Misalignments

Calculating SCF for Misalignment

The stress concentration factor for a misalignment as presented in Figure B-1, (ABS 1996), can be computed as follows:

$$SCF = 1 + 1.5\frac{e}{t_2} \tag{B-1}$$

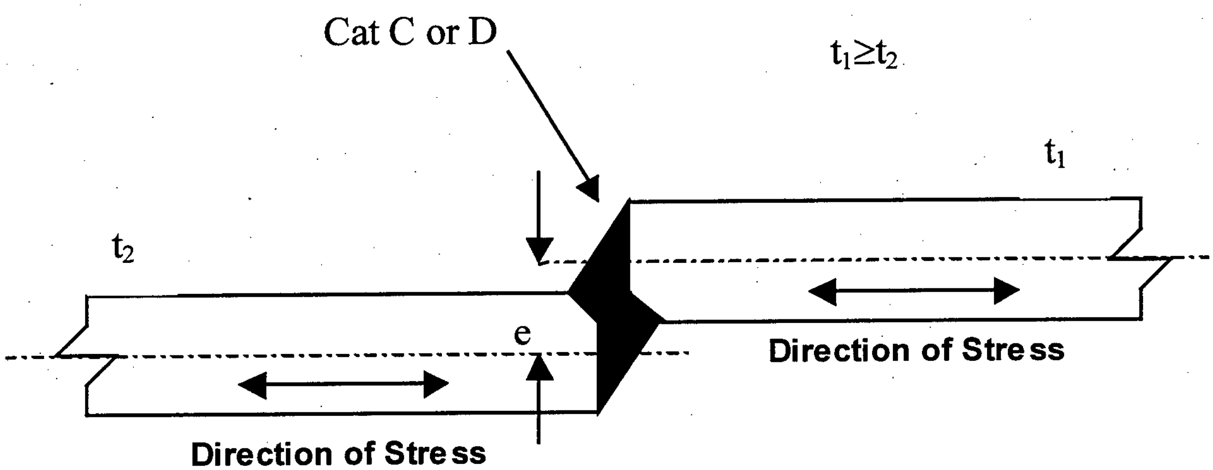


Figure B-1. Misalignment in Butt Welds, Full Penetration

Reference

ABS. 1996. *ABS Guide for Fatigue Assessment of Tankers*: American Bureau of Shipping.

Appendix C
Catalog of Structural Details

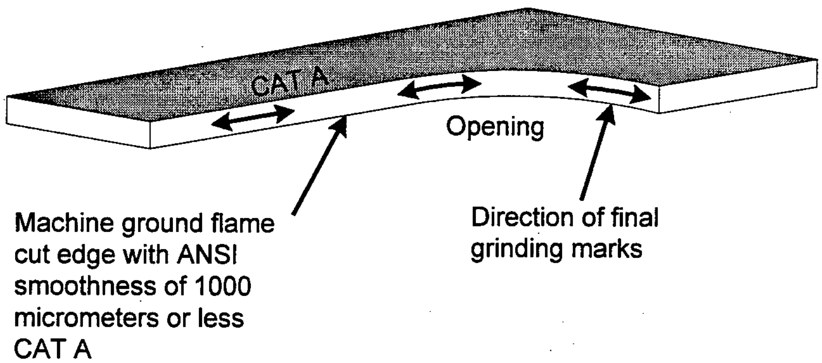


Figure C-1. Machine Ground Flame Cut CAT A

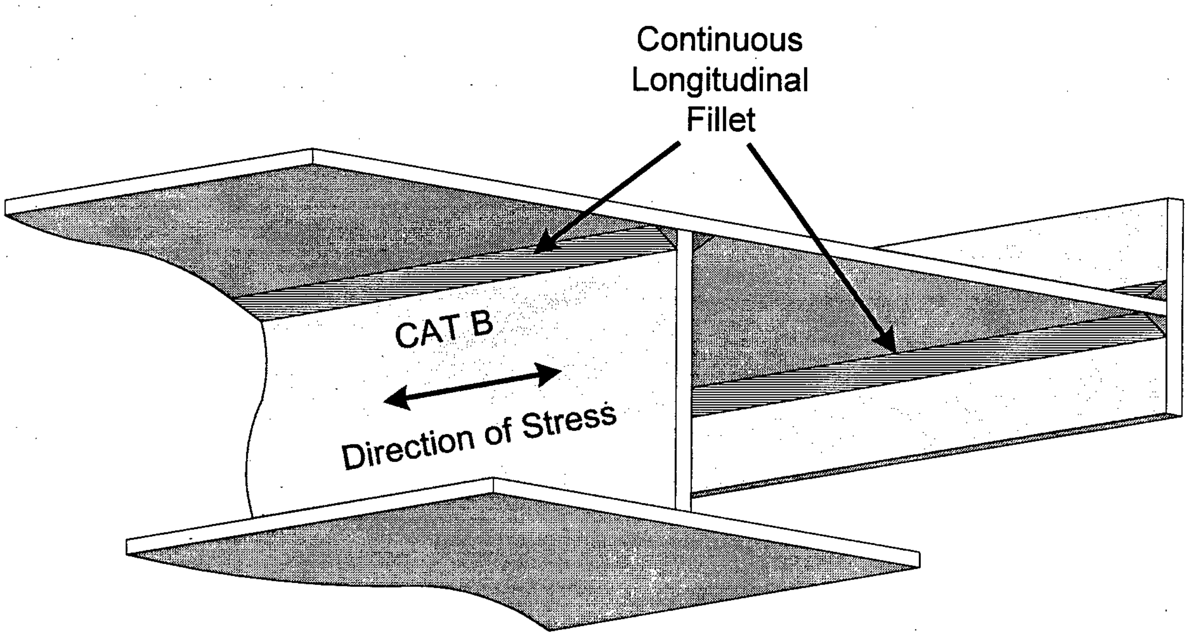


Figure C-2. Continuous Longitudinal Fillet Weld CAT B

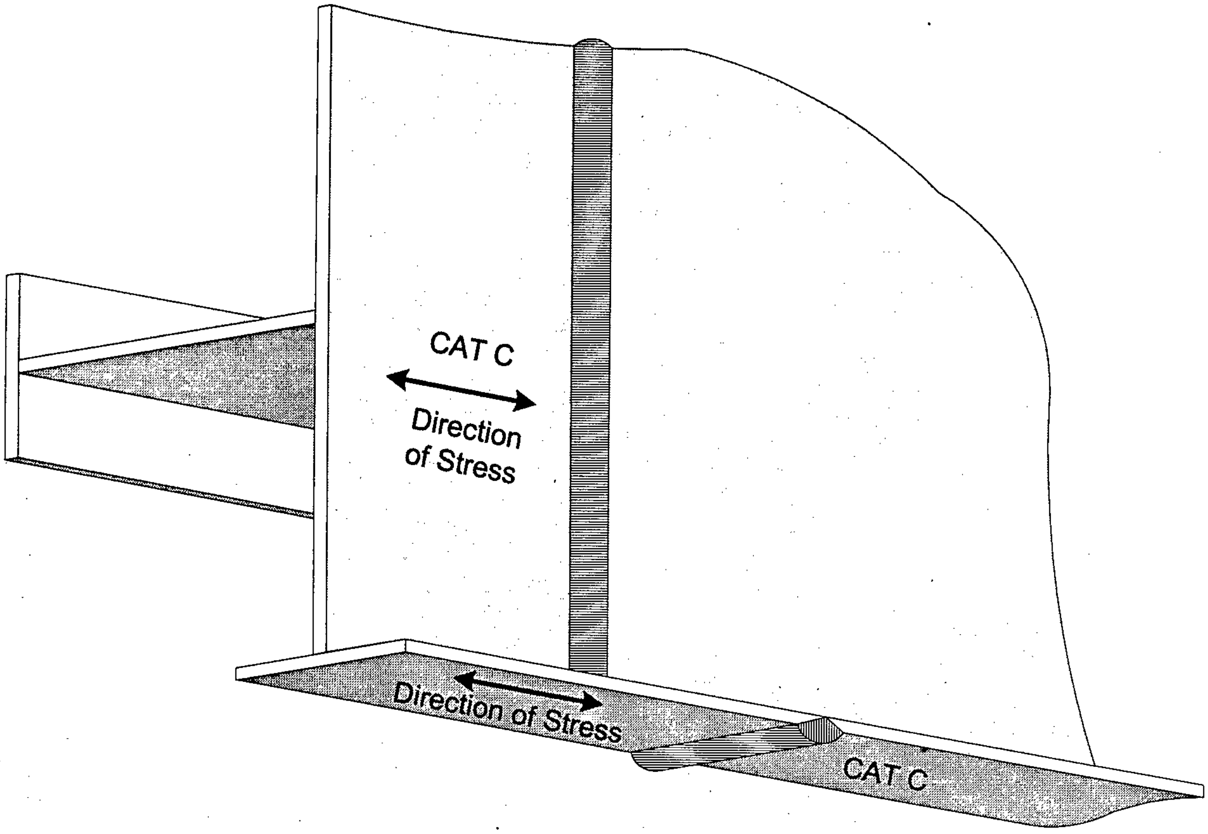


Figure C-3. Transverse Butt Weld CAT C

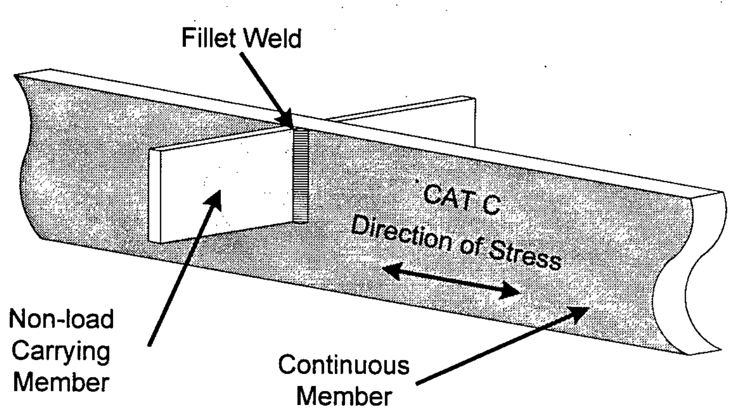
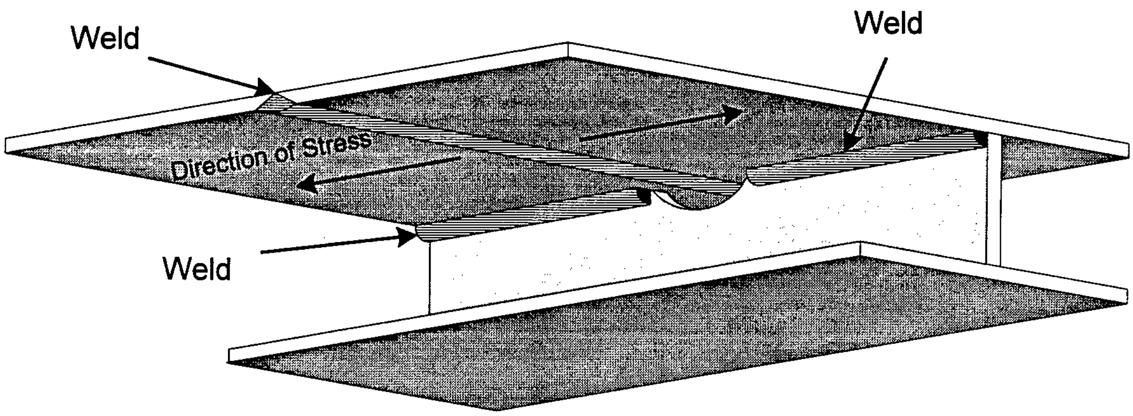


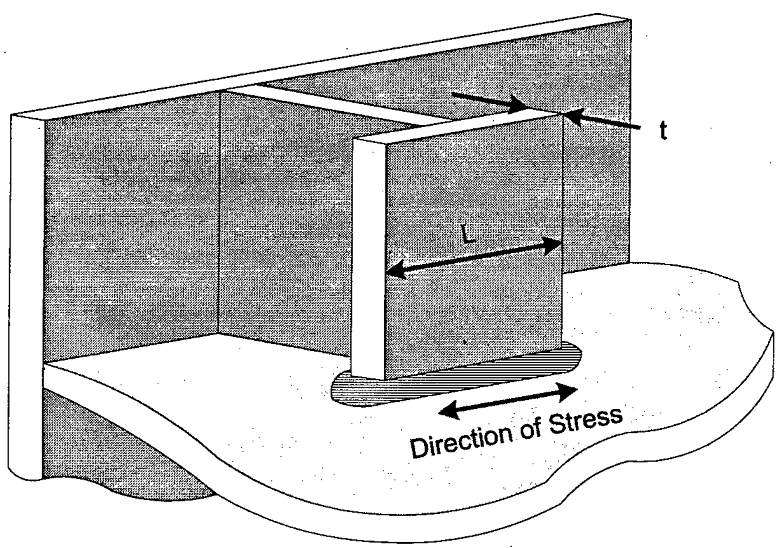
Figure C-4. Cruciform Joint CAT C



Rat Hole

CAT D:	Diameter < 4"
CAT E:	4" <= Diameter <= 6"

Figure C-5. Rat Hole CAT D or E



Non-load-carrying Attachment

CAT C:	L< 2"
CAT D:	2" <= L < 4" and L <= 12t
CAT E:	L > 4" or L > 12t and t < 1"
CAT E':	L > 4" or L > 12t and t >= 1"

Figure C-6. Non-Load Carrying Attachment CAT C, D,E, or E'

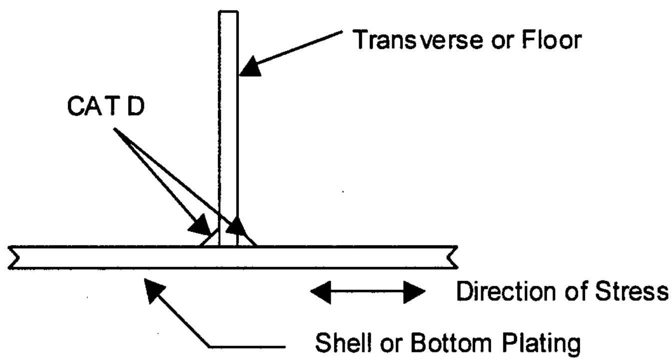


Figure C-7. Transverse Frame or Floor at Shell or Deck CAT D

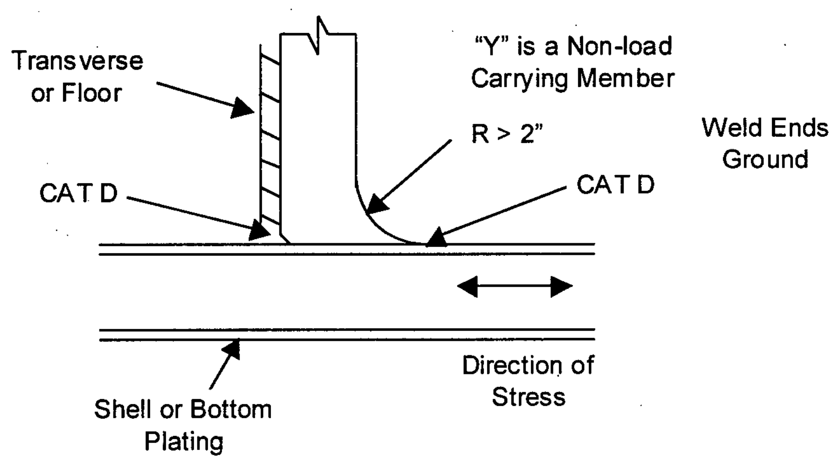
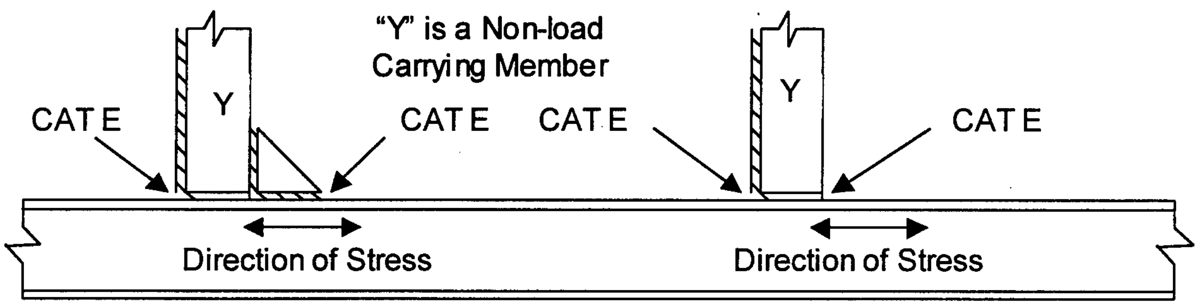


Figure C-8. Stiffener with Stress Parallel to Plane of Stiffener (with radius) CAT D



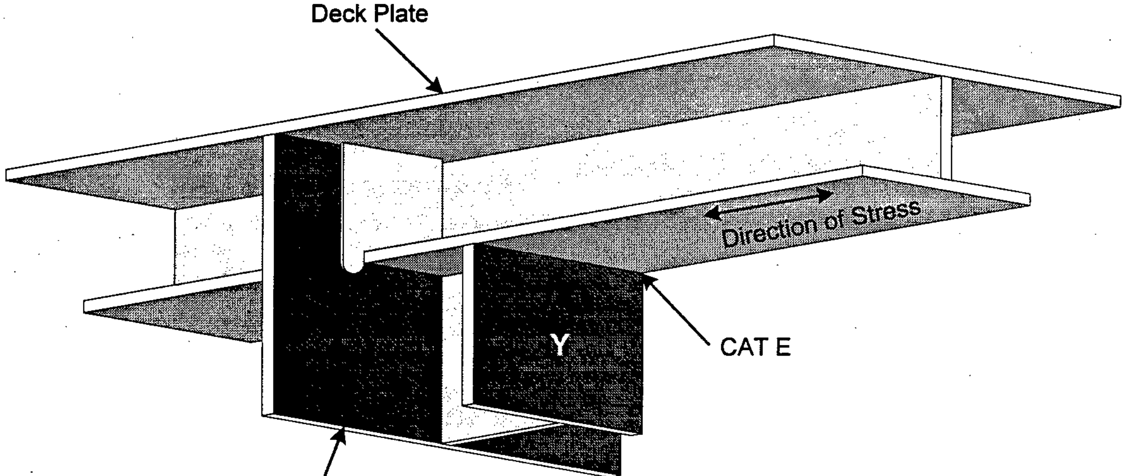
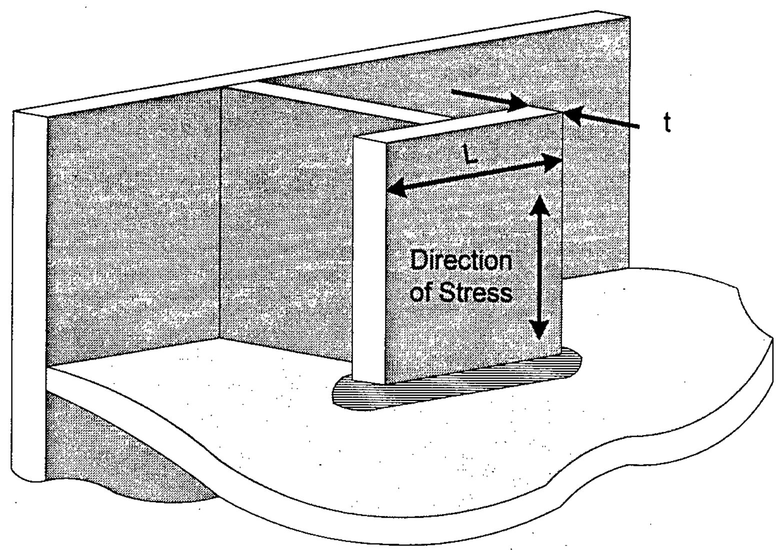


Figure C-9. Stiffeners with Stress Parallel to Plane of Stiffener (without radius) CAT E



Load-carrying
Attachment

CAT E: t < 1"
CAT E': t >= 1"

Figure C-10. Load Carrying Attachment CAT E or E'

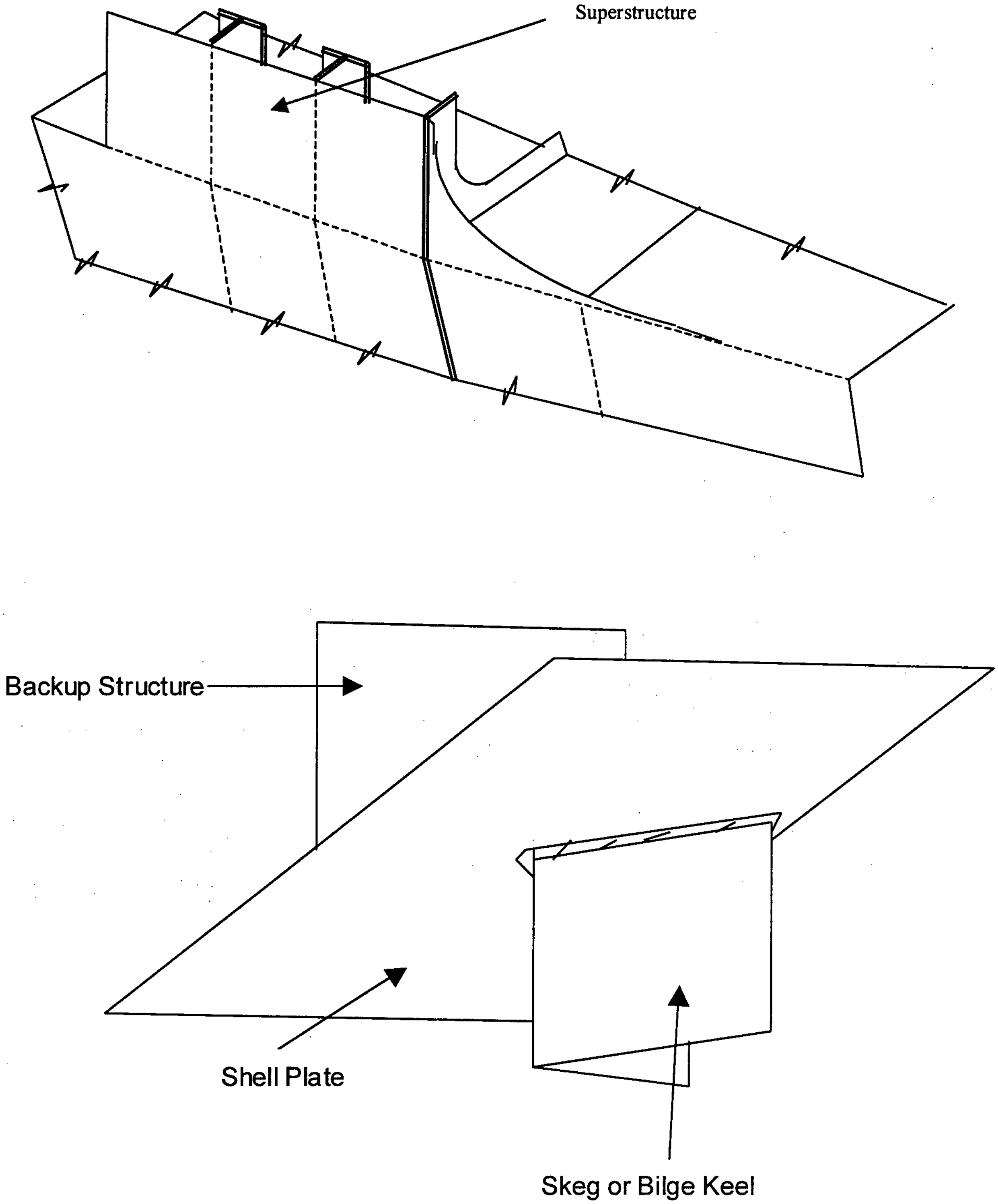


Figure C-11. Major Attachments that Require Finite Element Analysis

Appendix D
Example Computations

This appendix contains example calculations for nominal stress ranges, and stress concentration factors for openings.

Example 1 – Computing a Nominal Stress Range

The example presented in this section is provided in steps, and is for demonstration purposes only.

1. Obtain Ship Characteristics

Ship	DDG-2
Bow form	Fine
Ship service life	30 Years

L =	420 feet
B =	47 feet
Displacement	< 10000 Tons

2. Compute Hull-Girder Section Properties and Strength

Station	X (ft)	Section Modulus	
		Deck (in^2-ft)	Keel (in^2-ft)
0	0		
1	21		
2	42		
3	63	6705.48	5962.65
4	84	8954.40	7469.72
5	105	11102.94	8788.63
6	126	11983.04	9795.82
7	147	12662.37	9975.17
8	168	10915.38	11669.06
9	189	13701.60	13372.88
10	210	12504.78	12332.31
11	231	12039.13	11913.22
12	252	12219.02	11638.38
13	273	11963.59	10473.92
14	294	91997.01	9455.80
15	315	7598.27	7336.89
16	336	5990.96	5091.75
17	357	4719.78	4208.13
18	378		
19	399		
20	420		

3. Calculate the Maximum Bending Moment Range at Station 10:

Equation 20: BM_{max} [ft-tons] $= C1(L^{2.5}B)^{C2}$

Table 1: Hog: $C1 = 3.217 \times 10^{-4}$

$C2 = 1.038$

$BM_{max} = 3.217 \times 10^{-4}((420)^{2.5}47)^{1.038}$ [ft-tons]

$BM_{max} = 112{,}465$ [ft-tons]

Table 1: Sag: $C1 = 8.979 \times 10^{-5}$

$C2 = 1.113$

$BM_{max} = 8.979 \times 10^{-5}((420)^{2.5}47)^{1.113}$ [ft-tons]

$BM_{max} = 129{,}075$ [ft-tons]

Equation 21: BMR_{max} [ft-tons] $= BM_{max}(\text{hog}) + BM_{max}(\text{sag})$

$BMR_{max} = 112{,}465 + 129{,}075 = 241{,}540$ [ft-tons]

4. Calculate the Maximum Bending Moment Range at Station 0 – 20

For station 5:

Equation 22: $DF_5 = (1/2)(1 - \mathrm{Cos}(2\pi\, x_5 / L))$

$DF_5 = (1/2)(1 - \mathrm{Cos}(2\pi\ 105/420)) = 0.5$

Alternatively Table 2 could have been used to determine the *DF*

Equation 23: $BMR_{max\ 5} = DF_5 \times BMR_{max}$

$BMR_{max\ 5} = 0.5 \times 241{,}540 = 120{,}770$ ft –tons

Using the same procedure for all stations gives the following results:

Station	Maximum Bending Moment Range (ft-tons)
0	0
1	5,911
2	23,065
3	49,783
4	83,450
5	120,770
6	158,090
7	191,757
8	218,475
9	235,629
10	241,540
11	235,629
12	218,475
13	191,757
14	158,090
15	120,770
16	83,450
17	49,783
18	23,065
19	5,911
20	0

5. Calculate the Nominal Stress Range in Deck and Keel for Stations 0 – 20

For station 10:

$S_{10} = BMR_{max\ 10} \times 2.24/SM_{Deck}$

S_{10} = 241,540 ft-ltons× 2.24 kips/lton / 12,504.78 in^2-ft = 43.27 ksi

$S_{10} = BMR_{max\ 10} \times 2.24/SM_{Keel}$

S_{10} = 241,540 ft-ltons× 2.24 kips/lton / 12,332.31 in^2-ft = 43.87 ksi

Using the same procedure for all stations results in the following:

Station	Calculated Nominal Stress Range (ksi)	
	Deck	Keel
0	N/A	N/A
1	5.92 *	6.66 *
2	11.56 *	13.00 *
3	16.63	18.70
4	20.88	25.02
5	24.37	30.78
6	29.55	36.15
7	33.92	43.06
8	44.83	41.94
9	38.52	39.47
10	43.27	43.87
11	43.84	44.30
12	40.05	42.05
13	35.90	41.01
14	38.50	37.45
15	35.60	36.87
16	31.20	36.71
17	23.63	26.50
18	18.42 *	18.42 *
19	8.42 *	9.44 *
20	N/A	N/A

* For this example, linearly decreasing section modulus toward ends of ship was assumed. Actual values should be used in practice.

6. Determine the Permissible Stress Range:

The permissible stress range for the critical detail (category E) and 30-year service life, from Table 11, is 49 ksi.

7. Summary and Conclusions:

The permissible stress range exceeds the calculated nominal stress range in the deck and keel at all stations as shown in Figure D-1. Hull girder strength is adequate to prevent fatigue crack initiation.

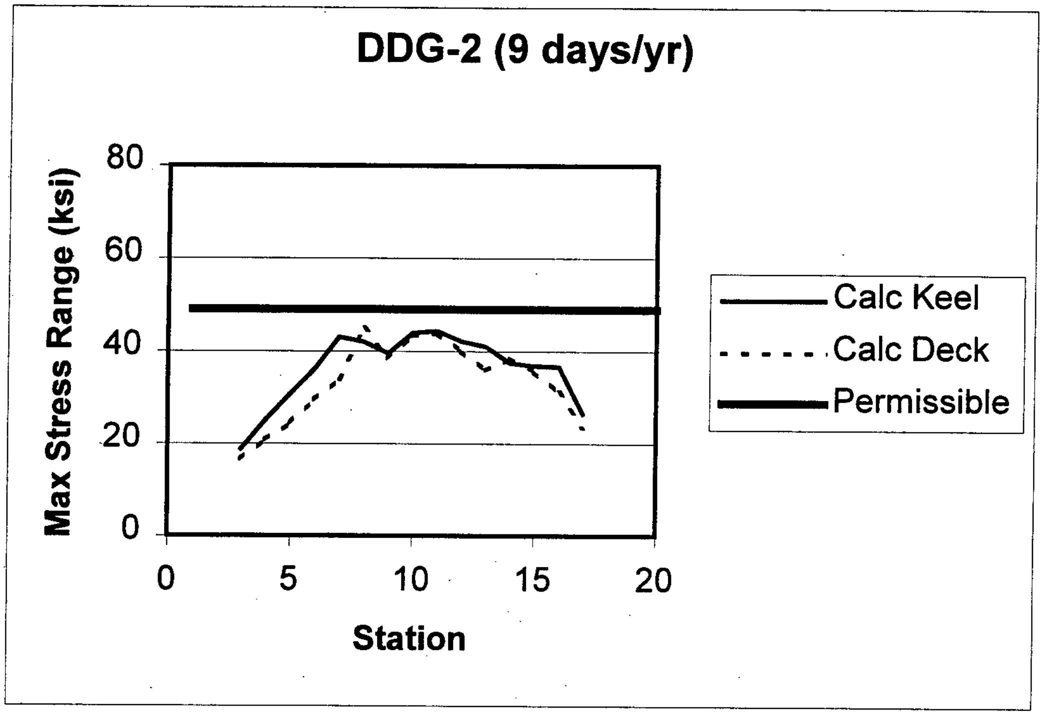


Figure D-1. DDG-2 Permissible Versus Computed Stresses

Example 2. Computing the Stress Concentration Factor for Opening

The example presented in this section is provided in steps, and is for demonstration purposes only.

1. Obtain Ship Characteristics

Ship: CG-16
$L =$ 510 feet

2. Opening Data

Opening:	H-01-82-2	$t_p =$	0.5 in
Located in HGE:	Yes	$t_i =$	0.5 in
$x =$	198.33 ft	$t_c =$	0.5 in
$B =$	610.6 in	$h =$	5.0 in
$b =$	30.0 in	$D =$	80.0 in
$a =$	48.0 in		
$r =$	7.5 in		

3. Calculate Basic SCF for Opening in Infinitely Wide Plate (Kb0)

Using $a/b = 48/30 = 1.60$

$r/b = 7.5/30 = 0.25$

Figure A-4 produces $K_{bo} = 2.55$

4. Calculate Basic SCF for Eccentric Opening in Finite Width Plate (Kb1)

Using Figure A-3:

$$S_1 = (B/2) - D - (b/2) = (610.6/2) - 80.0 - (30.0/2) = 210.3 \text{ in}$$

$$S_2 = B - S_1 - b = 610.60 - 210.30 - 30.0 = 370.30 \text{ in}$$

Equation A-4 yields

$$\alpha = K_{bo}\,[1/(1 + b/(2S_1))] + [1/(1 + (2S_1)/b)]$$

$$\alpha = 2.55\ [1/(1 + 30.0/(2 \times 210.30))]+[1/(1 + (2 \times 210.3)/30.0]$$

$$\alpha = 1.55[1/1.07133] + [1/15.020] = 2.44680$$

Substituting in Equation A-3 yields:

$$K_{b1} = \frac{\alpha[1 + b/(2S_2)](1 + b/S_1)^{0.5}}{1 - (S_1/S_2) + [(S_1/S_2)^2 + (S_1/S_2)(b/S_2)]^{0.5}}$$

$$K_{b1} = \frac{2.44680\,[1 + 30.0/2 \times 370.3)][(1 + 30.0/210.3)]^{0.5}}{1 - (210.3/370.3) + [(210.3/370.3)^2 + (210.3/370.3)(30/370.3)]^{0.5}}$$

$$K_{b1} = \frac{2.4460[1.04051]1.06895}{0.43208 + [0.60708]}$$

$$K_{b1} = 2.62$$

5. Calculate Stress Reduction Factor due to Reinforcing Ring (β)

Using

$r/b = 7.5/30 = 0.25$

$h/t_p = 5/0.5 = 10.0$

Figure A-5 yields

$\beta_0 = 2.0369\,(r/b)^3 - 1.7668\,(r/b)^2 - 0.044\,(r/b) + 0.7584$

$\beta_0 = 2.0369\,(0.25)^3 - 1.7668\,(0.25)^2 - 0.044\,(0.25) + 0.7584$

$\beta_0 = 0.67$

$t_c/t_p = 1.0$

$\beta_1 = 1.0$

Equation A-6 yields

$\beta = \beta_0\beta_1$

$\beta = 0.67 \times 1.0 = 0.67$

6. Reduce Stress Concentration due to Effect of Reinforcing Ring

Therefore, Eq. A-5 yields

$SCF_{max} = K_{b1}\,\beta$

$SCF_{max} = 2.62 \times 0.67 = 1.76$

7. Determine the Critical Detail and Category

Selecting an appropriate detail category can be based on examining its location. The reinforcing ring butt weld is located in an area of minimum stress; no attachment welded to ring. Critical detail is the longitudinal fillet weld that ties the coaming to the deck plate. Table 10 suggests classifying continuous longitudinal fillet weld as a category B detail.

8. Compare Calculated SCF to Maximum Permissible SCF

Table 12 provides maximum permissible stress concentration factors for both the extreme fiber of the ship and internal structure at the neutral axis. For a Category B detail, the maximum permissible SCF varies between 2.2 and 4.4, depending on whether the opening were located on the extreme fiber or on the neutral axis, respectively. Since the calculated maximum SCF, 1.76, is less than either of these values, the opening reinforcement is acceptable and the opening could be located anywhere on the ship, except in areas of the sheer and stringer strakes which require more detailed stress analysis due to the complex stress field.

Acronyms, Symbols, Definitions and Conversion Factors

Acronyms

DDS	design data sheets
FEM	finite element method
LRFD	load resistance factor design
NSWCCD	Naval Surface Warfare Center Carderock Division
SWATH	Small WAterplane Twin-Hull ship
USN	United States Navy

Symbols

A	= one of two empirical constants defining an S-N curve
B	= maximum beam of ship in feet
b	= one of two empirical constants defining an S-N curve
BM_{max}	= maximum lifetime bending moment
BMR_{max}	= maximum lifetime bending moment range
$C1$	= first of two factors defining the maximum bending moment amidships
$C2$	= second of two factors defining the maximum bending moment amidships
DF	= distribution factor
DF_i	= distribution factor at location x_i
FL	= fatigue life
g	= performance function
L	= length of the ship in feet, defined as the distance between perpendiculars LBP
L	= loads
N	= number of cycles to failure
n	= applied cycles for some stress range
P	= probability
P_f	= failure probability

R	= resistance or strength
SCF	= stress concentration factor
SL	= service life
SM_{Deck}	= section modulus of hull girder to deck
SM_{Keel}	= section modulus of hull girder to keel
S	= stress range (ksi)
x	= longitudinal distance from forward perpendicular (feet)
$S(\omega)$	= spectral density function
ω	= wave frequency
$R(\tau)$	= correlation function for a random process at any instant
τ	= time interval for two times t_1 and t_2, $\tau = t_2 - t_1$
$E(X^2)$	= mean square
RAO	= response amplitude operators
T_y	= life-time at sea
P_1	= ship heading probability
P_2	= ship speed probability
P_3	= wave height probability
P_4	= wave spectral probability
A_j	= area under an increment of the response function
ω_{iej}	= wave excited frequency of the ship at the ith mode and the j-th response
a	= crack size
ΔK	= $SY(a)\sqrt{\pi a}$, range of stress intensity factor
C, m	= crack propagation parameters
Y(a)	= function of crack geometry
Δ	= fatigue damage ratio
Δ_L	= limit on fatigue damage ratio
k_S	= fatigue stress uncertainty factor
f_i	= fraction of cycles in the ith block
k	= number of stress blocks in a stress (loading) histogram
Z	= section modulus

Definitions

Bottom Slamming	Impact generated when the bottom of a flat-bottomed bow emerges from the waves and rapidly reenters the water.
Bow Flare	Bow form that has large areas of plating which curve from being nearly vertical at the waterline to nearly horizontal at the strength deck. Wave flow is directed upward, and then outward, as the bow encounters waves. Aircraft carriers have this type of bow form.
Fine Bow	Bow form that has large areas of plating which are only slightly sloped from vertical. Bows of this type tend to cut through waves rather than slam onto them. Frigates, destroyers and cruisers typically have this type of bow form.
Flare Slamming	Impact generated when the flared portion of a large flared bow rapidly encounters waves and water rushing upward beneath it before being redirected outward.
Flat Bottom	Hull geometry where the side shell plating connects to flat horizontal plating beneath the ship.
Large Flare	Bow geometry having large areas of curvature upon which water tends to impact as the ship encounters waves and the flow is redirected from upward vertical to outward horizontal.
Lateral Bending	Athwartship bending of the hull girder about a vertical axis.
Operational Profile	The anticipated service conditions expected during the life of the ship broken down into amounts of time spent in a specific sea condition at a specific heading and speed.
Reliability	The probability that a component meets a specific performance requirement within some time period and under specified environmental conditions.
Ringing	The subsequent cyclic hull response which occurs after hull whipping. The response occurs at the lowest natural frequency of hull bending and is superimposed on the low frequency wave response.
Risk	The potential of adverse consequences commonly measured as a plot of occurrence probabilities and associated consequences.
Slam	Transient vertical bending moment excited along the length of the ship due to wave impacts.
Whipping	The initial high frequency response of the hull after a wave impact which magnifies the wave induced bending moments during the slam event. The magnitude and phasing of the whipping produces a dynamic response that tends to be larger in the sag direction than in the hog direction.

Conversion Factors

Load

units:	kN = kilo Newton
	lbs = pounds
	Lton = long ton
	Mton = Metric ton
1 Lton	= 2240 lbs
1 kip	= 1000 lbs
1 kip	= 4.448 kilo Newtons
1 Mton	= 2205 lbs

Stress

units:	ksi = kips per square inch
	tsi = tons per square inch
	Mpa = mega-Pascal
1 ksi	= 1000 psi
1 ksi	= 6.895 Mpa
1 tsi	= 2.24 ksi

Length

units:	cm = centimeter
	ft = foot, feet
	in = inch
	m = meter
1 in	= 2.54 cm
1 ft	= 0.3048 m

Speed

units:	ft/s = feet per second
	kt = knot
	m/s = meters per second
1 kt	= 1.688 ft/s
1 kt	= 0.5145 m/s